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RESEARCH AND DEVELOPMENT OF MATERIEL

ENGINEERING DESIGN HANDBOOK

CARRIAGES AND MOUNTS SERIES

ELEVATING MECHANISMS

HEADQUARTERS, U. S. ARMY MATERIEL COMMAND DECEMBER 1964

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AMCP 706-346, Elevating Mechanisms, forming part of the Carriages and Mounts Series of the Army Materiel Command Engineering Design Handbook Series, is published for the information and guidance of all concerned.

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PREFACE

The Engineering Design Handbook Series of the Army Materiel Command is a coordinated series of handbooks containing basic information and fundamental data useful in the design and development of Army materiel and systems. The handbooks are authoritative reference books of practical information and quantitative facts helpful in the design and development of Army materiel so that it will meet the tactical and the technical needs of the Armed Forces.

This handbook has been prepared as one of a series of eight handbooks on Carriages and Mounts. It presents information and data on the fundamental operating procedures and design of Elevating Mechanisms.

This handbook was prepared by The Franklin Institute for the Engineering Handbook Office of Duke University, prime contractor to the U. S. Army Research Office-Durham. Technical assistance and guidance were rendered by the Army Weapons Command and the Army Tank-Automotive Center.

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Comments and suggestions on this handbook are welcome and should be addressed to Army Research Office-Durham, Box CM, Duke Station, Durham, North Carolina 27706.

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α = maximum elevating acceleration

α_b = deceleration of tipping parts induced by buffers

α_{br} = deceleration at brake

α_{eb} = braking deceleration of tipping parts

α_p = pitching acceleration of vehicle

α_{xx} = acceleration of gear unit, general expression

β = gear pressure angle; angle between buffer and buffer stop

Υ = ratio of motor speeds, maximum operating to peak

η = gear train efficiency

η_g = efficiency of each meshing gear

η_w = efficiency of worm gear

θ = angle of elevation

θ_b = angular buffing distance

θ_{bm} = maximum angular buffing distance

θ_{br} = brake drum travel

λ = lead angle of worm gear, helix angle

μ = coefficient of friction, general expression

μ_b = coefficient of friction of thrust bearing

ν = Poisson's ratio

ρ = mass density of hydraulic fluid

σ = general expression for stress

σ_e = endurance limit

Φ = mass moment of inertia of tipping parts about trunnions

Φ_e = effective mass moment of inertia of elevating system

Φ_{xx} = mass moment of inertia of gear unit, general expression

ω = elevating velocity

ω_b = angular velocity of tipping parts during buffing

ω_{br} = angular velocity of brake

ω_m = motor speed

a_o = orifice area

A_b = effective area of buffer piston; surface area of brake shoe

A_c = surface area of clutch

A_e = effective pressure area of hydraulic elevating mechanism

c_o = orifice coefficient

c_v = Barth's velocity factor

C_F = ratio, face width to circular pitch

C_i = gear tooth inbuilt factor

d_g = groove depth of orifice

D = brake diameter

D_b = effective diameter of thrust bearing

D_p = pitch diameter

D_w = pitch diameter of worm

E_a = energy absorption rate

E_b = energy absorbed by brake

E_r = energy rate

F_a = inertia force of recoiling parts

F_b = buffer force

F_{ba} = applied brake force

F_{ca} = applied clutch force

F_e = effective face width of gear; applied force of hydraulic elevating mechanism

F'_e = effective force of hydraulic elevating mechanism

F_g = propellant gas force; gear tooth load

F_{gs} = limiting gear tooth load for strength

F_{gw} = limiting gear tooth load for wear

F_N = force normal to surface

F_R = reaction of buffer stop

F_s = spring load

F_T = trunnion load

F_w = face width of gear tooth

F_{xx} = gear tooth load, general expression

F_α = force on tipping parts induced by pitching acceleration

F_ϕ = normal load on buffer rod end

HP = horsepower

K = recoil force

K_w = wear factor of gear tooth

LIST OF SYMBOLS (Concluded)

L = subscript denoting left brake shoe
L_b = effective length of buffer rod
L_e = length of hydraulic elevating mechanism
M_e = equilibrator moment
M_f = brake shoe frictional moment
M_F = applied brake shoe moment
M_N = brake shoe pressure resisting moment
M_w = weight moment
n = number of gear meshes in train
N = number of gear teeth
N_w = number of threads in worm
p_c = clutch contact pressure
p_e = pressure in hydraulic elevating mechanism
p_m = maximum pressure on brake band
P_c = circular pitch
P_d = diametral pitch
P_L = axial pitch of worm
P_m = power generated by motor
P_t = power required to rotate traversing parts
Q = flow rate
r_b = bearing radius
r_g = gear train ratio
r_n = ratio of peak speeds, motor to mount
r_t = radius, CG tipping parts to trunnion axis
R = subscript denoting right brake shoe; radius from trunnion to pivot on cradle
R_b = radius, trunnion axis to line of action of buffer
R_{px} = pitch radius of gear, general expression
R_v = radius, CG vehicle to trunnion axis
R_ϕ = radius, buffer contact point to trunnion axis
R_μ = radius of friction circle
t_b = braking time
t_m = time constant of motor
t_t = time constant of tipping parts
T = torque about trunnions induced by buffers
T_a = accelerating torque of tipping parts
T_b = frictional torque of trunnion bearings; braking torque
T_c = torque transmitted by clutch
T_e = residual weight moment after equilibration
T_E = torque on elevating arc
T_f = firing torque
T_m = torque at power source
T_t = torque induced by secondary recoil acceleration
T_w = torque on worm gear
T_{xx} = torque on gear unit, general expression
T'_{xx} = accelerating torque of each gear unit, general expression
T_α = accelerating torque of gear train
$T_{\alpha b}$ = torque induced by buffers to stop mass of elevating system
v_b = linear velocity of buffer
v_e = linear velocity of hydraulic elevating mechanism
v_p = pitch line velocity of gear
v_t = tangential velocity at end of turning radius
w = brake band width
w_c = width of clutch
W_t = weight of tipping parts
x = subscript denoting gear number, general expression
x_b = buffer stroke at any position θ_b
xx = subscript denoting gear unit, general expression
y' = gear tooth form factor, dynamic load

CARRIAGES AND MOUNTS SERIES
ELEVATING MECHANISMS*

I. INTRODUCTION

A. PURPOSE

1. This is one of a series of handbooks dealing with various components of Carriages and Mounts. The material in this handbook presents procedures for the design of elevating mechanisms. Various types of mechanisms including their operating characteristics are discussed. Figure 1 shows one type of elevating mechanism assembled to the mount. This handbook is devoted exclusively to elevating mechanisms except where limited descriptions of other carriage components are needed to show congruity among these components. More elaborate descriptions of these components appear elsewhere in the Carriages and Mounts Series. Elevating and traversing mechanisms have many similar characteristics and requirements and use similar machine elements, therefore, to avoid excessive cross-referencing, some of the material in this handbook is duplicated in ORDP 20-347, *Traversing Mechanisms*† This text should not be considered the ultimate in design concepts nor should designers always adhere strictly to its suggestions. On the other hand, new personnel should find this handbook a helpful guide in becoming familiar with the purpose, functions, and design aspects of elevating mechanisms while experienced personnel should find it a useful reference.

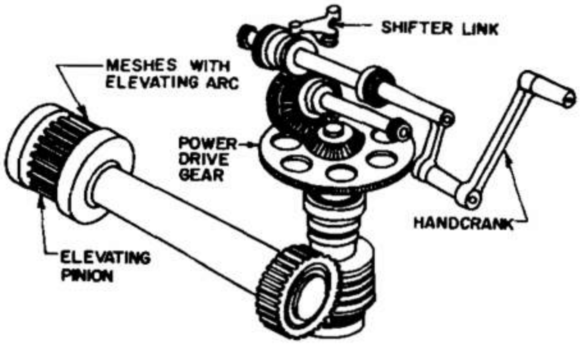


Figure 1. Power Elevating Mechanism

B. FUNCTIONS

2. The long range of projectiles and missiles requires extremely accurate aiming. Slight errors in the pointing of either cannon or launcher ramp can result in missing the target by wide margins. When aimed by direct sighting, a weapon must be moved slowly and precisely to align it accurately with the target. When aimed by a separate fire control unit the cannon or the missile launcher must be able to respond accurately to the direction signals of the unit. In either case, such a weapon is too heavy to be aimed directly by hand. Hence, a handwheel or power operated mechanism is provided which helps a gunner to attain a precise position and hold it there during firing. On the other hand, many small arms are light enough to be aimed without the aid of a mechanism. Position coordinates are given in angles about two axes. Normally one coordinate is in elevation about a horizontal axis and the other is in traverse about a vertical axis.

3. The angle of elevation is the included angle between the bore axis and the horizontal. This angle, subject to the ballistics and the relative positions of gun and target, determines the length of the trajectory and hence the range. To assume the correct angle of elevation, the tipping parts and therefore, gun tube, must be rotated about the cradle trunnions. The elevating mechanism is the apparatus which imparts this rotation, controls it, and firmly holds the tipping part during firing. Torques about the trunnions arising from the eccentricities of inertia and retarding forces during recoil, or from any other source, are transmitted to the top carriage by the elevating mechanism. It is true that the muzzle preponderence caused by the muzzle-wise position of the center of gravity of

* Prepared by Martin Regina, Laboratories for Research and Development of The Franklin Institute.

† Reference 1. References are found at the end of this handbook.

the tipping parts is very nearly balanced by the equilibrator.* However, equilibration is not perfect under all conditions, especially during recoil when a fluctuating torque induced by recoil dynamics must be resisted by the elevating mechanism.

4. The elevating mechanism and its controls must be designed for easy operation to the extent that the gunner can devote most of his attention to the target. An optimum mechanism must combine precision, reliability, durability, speed and low-power demand, with ease of control. For some installations, it must develop high acceleration, yet must be able to stop smoothly without overrun. Essential mechanical characteristics of elevating mechanisms on missile launchers are similar to those on guns. Although they are generally subjected to less severe loads, they must be shielded from the rocket blast, or the structure and materials used must be capable of withstanding the blast.

II. TYPES OF ELEVATING MECHANISM

5. Elevating mechanisms are essentially gear trains or linkages; one terminal at the power source, the other on the tipping parts. The gear train frequently includes a self-locking worm and worm wheel, otherwise a mechanical brake must be provided to hold the tipping parts in the prescribed position. The manually operated type and the power operated types form the two general categories. The various subtypes are named according to the method of construction or assembly and may often fall under both general groups. Manually operated units are installed on those weapons which do not demand elevating activity at high torques, at high speeds, or for prolonged periods. Effort is applied by a handwheel. If elevation is too burdensome, the mechanism must derive its power from mechanical or electrical sources. However, in case of power failure, these mechanisms are geared for handwheel operation should such an emergency occur.

A. SCREW AND NUT TYPE

6. Probably the simplest mechanism is the screw and nut type. This type, readily adaptable to the smaller guns, is self-locking and has an inherently high mechanical advantage but the range of elevation is limited to relatively small angles. Two versions are available. Figure 2(a) has a captive nut pivoted to the cradle. Elevation is achieved by turning the screw through handwheel and gear train, causing the nut to travel in the desired direction. The base of the screw rests in a thrust bearing whose housing pivots on the top carriage. Two bevel gears serve as drivers for the screw. One is fixed to the screw near its lower end. The other rotates on the same axis as that of the bearing housing pivot, so no matter what angle is assumed by the elevating screw, a proper gear mesh is assured. The second version operates similarly. It differs by having the elevating screw pivoted to the cradle with the nut free to turn and tilt (Figure 2(b)).

B. PINION AND ARC TYPE

7. This type may be labeled conventional. It is adaptable to all types and sizes of weapons. It may be operated manually or mechanically. It may be self-locking or have a brake. Like the screw and nut type the pinion and arc type has two versions, both identical except for the last two members of the gear train, the pinion and elevating arc. Figure 3(a) represents one method of installation. It has the mechanism attached to the top carriage and elevating arc attached to the cradle. Change in elevation occurs when the pinion rotates the elevating arc about the trunnions. The second method has the arrangement reversed (Figure 3(b)). Here the arc is attached to the top carriage while the mechanism is fixed to and rides with the cradle. This second version, because of additional mass and installation difficulties, is not particularly adaptable to power drives if some or all power equipment are included with the tipping parts.

C. HYDRAULIC TYPE

8. An elementary version of the hydraulic elevating mechanism exploits the existing equilibrator for its functional needs merely by adjusting the length of equilibrator stroke. The equilibrator may be either mechanical spring or hydropneumatic. The top carriage end of the equilibrator is filled with oil or some other hydraulic fluid. A displacement pump mounted on either top carriage or cradle, contains another volume of oil.

* Reference 2.

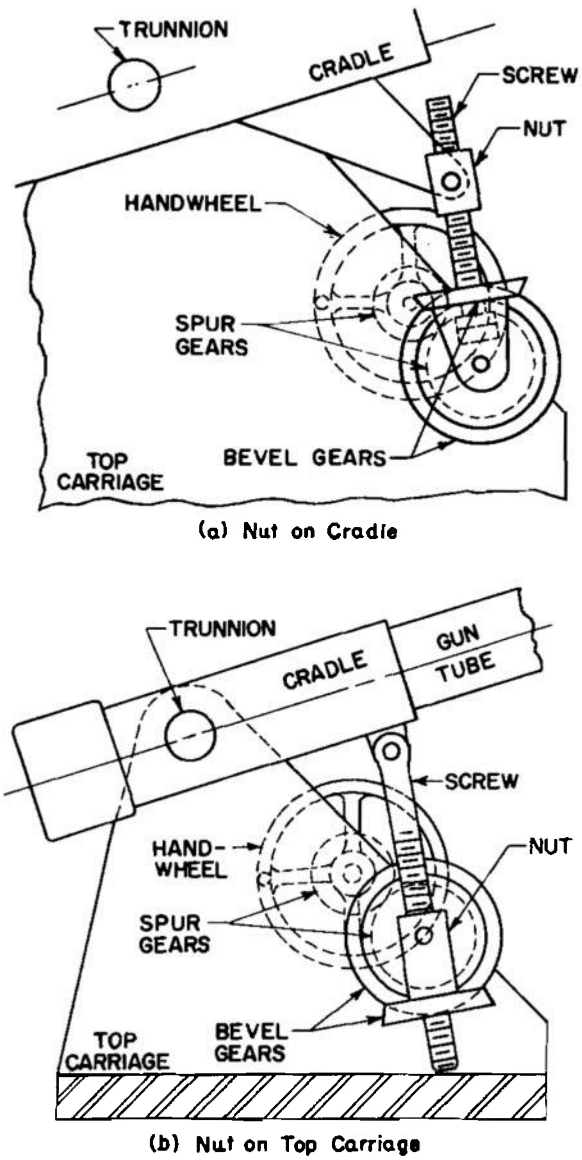


Figure 2. Screw and Nut Type Elevating Mechanisms

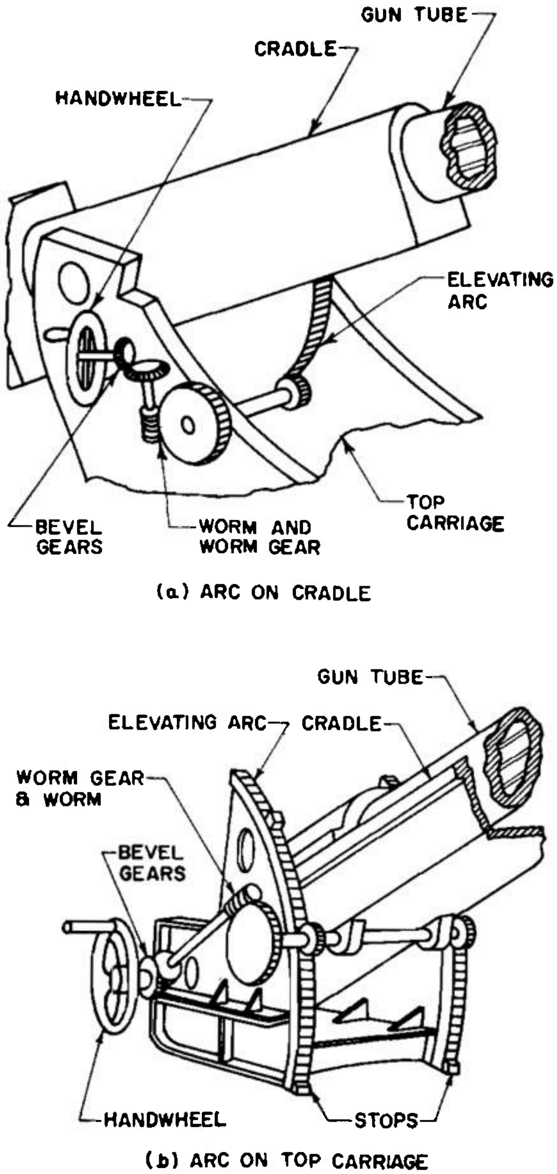


Figure 3. Pinion and Arc Type Elevating Mechanisms

The two oil chambers are connected by a flexible tube. This oil actually is an adjustable rigid link between pump and equilibrator. In this type, overequilibration is essential to maintain this rigid hydraulic link at all angles of elevation. Figure 4 illustrates the mechanics of operation. Angle of elevation corresponds with pump piston setting which is adjusted by handwheel or other power source. As the oil moves toward the equilibrator, effectively lengthening its stroke, the angle of elevation decreases. As the piston is withdrawn, oil flow reverses leaving to the equilibrator the task of taking up the slack created by the evacuated oil chamber and moving the tipping parts to a higher angle. Motion ceases when the oil link once again becomes rigid.

9. An example of a more elaborate hydraulic elevating mechanism for power actuated tank guns is schematically represented by Figure 5. This type is a hydraulic strut which readily lends itself to power elevation by a positive displacement pump which forces the required volume of oil into either end of the strut to affect the desired change in elevation. Flow is constant at pressures varying within the capacity of the pump. These pressures are regulated by the demand, *i.e.*, whatever is needed to overcome the frictional and inertial resistances of the tipping parts. In keeping with other power elevating mechanisms, this one is also equipped with a hand pump for manual operation during emergencies. The mechanism may be colinear and integral with the equilibrator or it may be a separate unit. In either case it may be equipped with a shock absorber to hold the gun in position when subjected to the inertial loads of traveling and firing. Figure 5 represents a unit comprising elevating mechanism and shock absorber. Outlining its response to the elevating

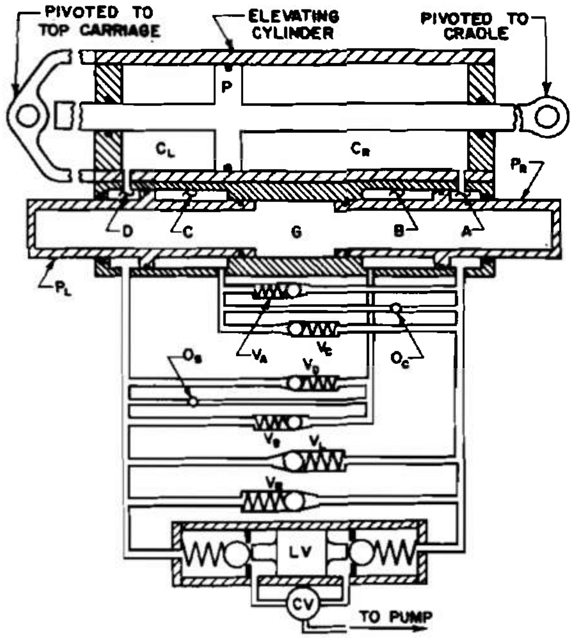


Figure 5. Hydraulic Elevating Mechanism—Power Type With Shock Absorber (Schematic)

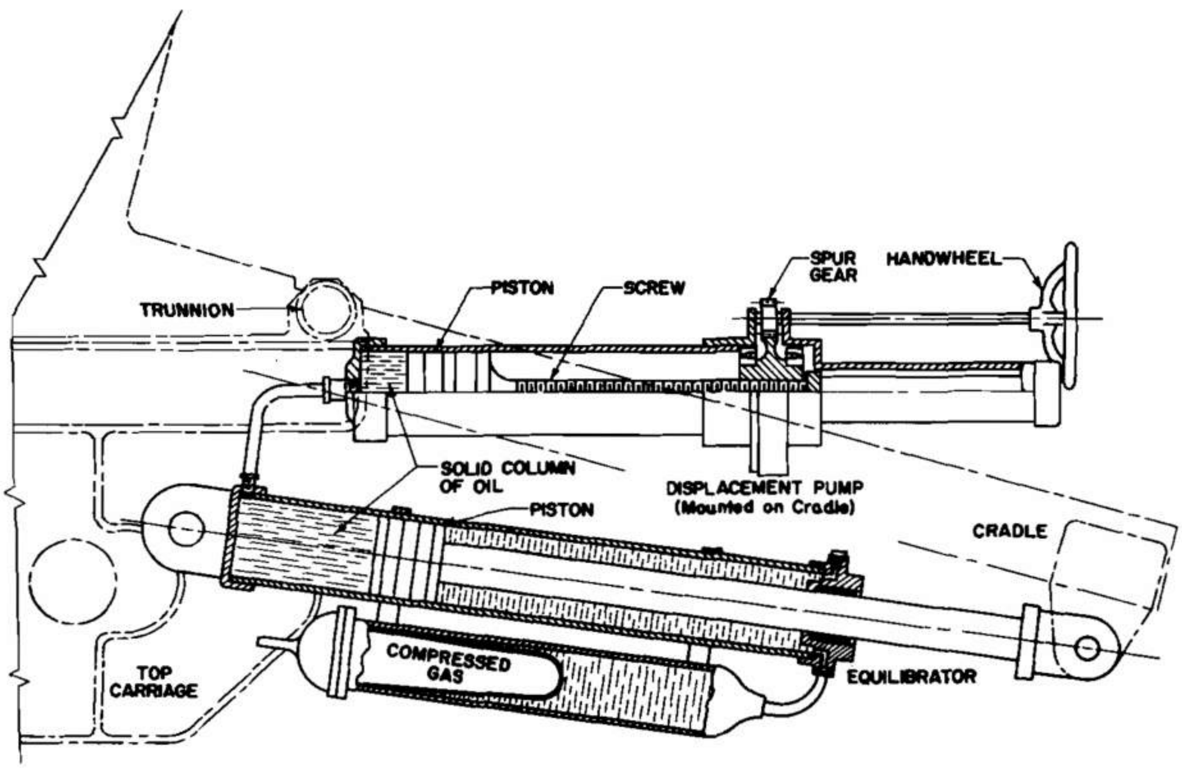


Figure 4. Hydraulic Elevating Mechanism—Manual Type (Combined With Equilibrator)

piston motion in sequence illustrates the functions of the shock absorber.

The symbols in Figure 5 and in the discussion follow.

A,B,C,D = chambers of shock absorber
A_A,A_B,A_C,A_D = annular pressure areas of shock absorbers (equal values)
A_L,A_R = area of shock absorber pistons, gas side (equal areas)
A_P = area of elevating piston, either side
C_L,C_R = left and right elevating cylinder chambers
CV = control valve
G = gas chamber of shock absorber
LV = locking or check valve
O_C = orifice in line between chambers A and C
O_B = orifice in line between chambers B and D
p_A,p_B,p_C,p_D = oil pressure in chambers A, B, C, D
p_G = gas pressure
p_L,p_R = oil pressure in left and right elevating cylinders
p_1 = rated pressure of valves V_B and V_C
p_2 = rated pressure of valves V_A and V_D
p_3 = rated pressure of valves V_R and V_L
P = elevating piston
P_L,P_R = left and right shock absorber pistons
V_A,V_B,V_C,V_D,V_L,V_R = relief valves

Comparative pressure values are assigned for convenience.

$$p_1 = \frac{1}{2}\frac{A_R}{A_A}\,p_G \text{ where } A_R > A_A$$

$$p_2 = 2\,p_1$$

$$p_3 = 5\,p_1$$

10. Elevation is achieved by pumping oil under pressure from either power or hand pump through chamber A to cylinder C_R or, to reverse direction, through chamber D to cylinder C_L. The locking valve provides a free fluid channel during pumping but closes automatically when pumping ceases, thus precluding oil feed back and therefore reverse rotation of motor or handwheel. During normal elevating performance, assuming 100% efficiency,

$$p_R = p_A < p_1$$

$$p_A A_A < p_G A_R$$

11. When normal elevating activity is suspended and forces induced by the pitching acceleration of the tank are present, the shock absorber becomes active. As pistons P_R and P_L are limited in their travel (in extreme outward positions shown in Fig. 5), no appreciable pressure derived from p_G is transmitted to cylinders C_R and C_L and $p_A = p_D \approx 0$. If the accelerating force on P is to the right, it induces a pressure p_R in C_R and subsequently p_A in A. Should $p_A A_A < p_G A_R$, the shock absorber piston, P_R, will not move. Neither will the oil flow into C because that chamber is already full, nor into D because $p_A < p_3$ and V_R will not open. Therefore P will experience no motion other than that provided by the compressibility of the oil. Should the load be large enough so that

$$p_3 > p_A > \frac{A_R}{A_A}\,p_G + p_B$$

P_R will move to the left forcing oil from B into D and eventually into C_L to fill the void caused by the displacement of P when oil from C_R flows into A. The pressure, p_B, varies from zero to p_1 depending on the oil velocity through O_B. If pitching velocities are large, the restriction at O_B will impede flow sufficiently to raise the pressure to p_1 which will open valve V_B. Thereafter, p_B will remain constant until P_R seats. The total travel of P_R is equivalent to a small angular displacement of the tipping parts, on the order of 3 to 4 degrees. If pitching ceases at any time while P_R is moving to the left, the gas pressure will force the tipping parts to their original position as the oil reverses its flow from C_L and O_B into B. On the other hand, should pitching accelerations continue in the same direction after P_R seats, P_A will reach p_3 to open valve, V_R, thereby permitting piston, P, to continue its motion with a subsequent change in elevation. But this additional change in elevation cannot be compensated for by the shock absorber, its compensating ability being confined to the travel of P_R. Therefore, the inadvertent elevation beyond the control of the shock absorber can be corrected only by regular elevating procedures.

12. Assume that the right piston, P_R, has just completed its stroke to the left and that pitching acceleration reverses quickly before the gas pressure has time to return all moving parts to their original positions. The elevating piston, P, immediately begins to move to the left under the combined influence of tipping parts and shock absorber gas pressure pushing P_R to the right. The oil from C_L is now forced to flow into chamber B, its only restriction to flow being at O_B. When P and subsequently, P_R tend to move too fast for O_B to cope with an equivalent flow needed to replenish B, further resistance is realized until p_D eventually reaches p_2 and opens valve, V_D. At this stage p_2 is still less than $p_G\, A_L/A_D$. After V_D opens, the oil pressure will remain constant until P_R becomes seated on the right, and if pitching continues to move P to the left, the sequence of shock absorber response will parallel that previously described but this time its left components will regulate the activity.

This model shock absorber has been discontinued, in favor of simpler mechanisms, such as that represented in Figure 6. These mechanisms are rigid and strong enough to receive the shock loads caused by pitching accelerations with no appreciable displacement of the tipping parts.

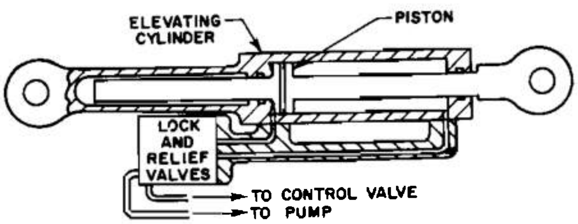


Figure 6. Hydraulic Elevating Mechanism—Power Type Without Shock Absober

13. The greatest advantage offered by the hydraulic elevating mechanism is its compactness. It requires only a small space, making it ideal for self-propelled weapons where space is at a premium. Combined with the advantages of the symmetrical weight distribution and the compactness offered by the concentric recoil mechanism, this advantage far outweighs the minor disadvantages. One of the latter appears in the type mechanism of Figure 4 where the weapon must be overequilibrated for elevating. Overequilibration in itself is not serious but it must be remembered that the vertical firing couple, if not small or if not counteracted by the inflexible oil link, will depress the tipping parts and jeopardize the weapon's accuracy. The mechanics of the firing couple, T_f, are illustrated in Figure 7. If trunnions and mass center of the recoiling parts lie on the bore axis, no firing couple can develop. This is the ideal arrangement and every design facet should be explored to achieve it. If the trunnions do not lie on the bore axis, firing couples will occur throughout the recoil cycle. The couple will oppose equilibration when the trunnions are above the axis but will assist when they are below. Occasions will arise when colinear forces cannot be arranged, thereby yielding firing couples of appreciable magnitude even though the offset may be small. These couples must be transmitted through the elevating mechanism.

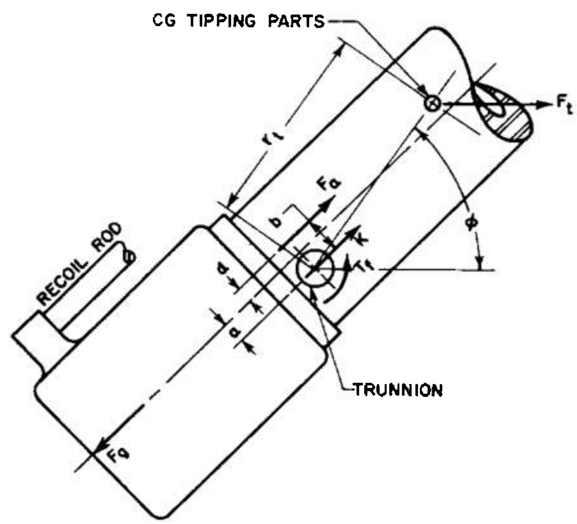


Figure 7. Firing Couple Diagram

Leakage can be another deterrent to the proper functioning of the hydraulic elevating mechanism. If oil leaks from the elevating mechanism, the pump piston must be repositioned to preserve the solid oil link. Loss of oil here means that minimum angles of elevation cannot be attained. Leakage in the equilibrator will render all elevating activity impossible. Although leakage can be serious, present design techniques have reduced it to a minor problem readily corrected by routine maintenance practices.

III. EQUIPMENT ASSOCIATED WITH ELEVATING MECHANISMS

A. EQUILIBRATORS

14. The equilibrator is perhaps the best crutch available to the elevating mechanism. As a counterbalance for the tipping parts, it reduces the physical effort needed for elevating to little more than that required to overcome friction, making manual elevation possible for even the largest guns without resorting to gear trains of prohibitively high gear ratios. Not only does it aid manual effort, it also aids mechanically powered elevating systems by reducing power effort to reasonable limits. It is an indispensible component of the hydraulic elevating mechanism. In short, it provides the static force which enables the elevating mechanism to function with the least amount of effort. Getting rid of gravity moment bias by equilibration enables designing for better transient response in tank, airborne, or AA gun following a moving target by radar director than otherwise would be possible, no matter how lavish a power consumption is permitted. Equilibration is a symbol of excellent problem reduction approach to a well-designed weapon. The discussions on the various phases of equilibrator design are found in Reference 2.

B. POWER DRIVE

15. Quick-elevating heavy guns require a sustained effort far beyond the stamina of anyone operating a manual elevating mechanism. This is especially true of high-rate-of-fire antiaircraft weapons which must keep pace with fast and elusive targets and of tanks which are constantly moving, therefore contending with changing target positions. Such mounts are elevated with variable speed power drives which are generally mounted on the top carriage.

Power drives, capable of both coarse and fine elevation or depression, are electric or hydraulic. All power for a mobile or self-propelled weapon is normally derived from an internal combustion engine. (For permanent emplacements, conventional electric transmission lines may be available.) The engine may drive either a generator or a hydraulic unit directly. If a direct drive is used, only one mount can be served but with a motor-generator unit, this number may be increased. The engine may be attached to the mount, it may be a separate unit and borne on its own chassis, or it may be that of the prime mover equipped with a power take-off device. When rough control can be tolerated, coarse elevation can be achieved by gearing the engine directly to the elevating mechanism. However, this arrangement will not be suitable for fine elevation until quick-operating and sensitive controls become available for it.

16. Hydraulic drives are much more refined systems than the direct engine drive and are readily adapted to precise automatic control. The system features a hydraulic pump and motor (Figure 8). One such system has a variable-displacement pump with a direct line to the motor. Another has a constant-displacement pump discharging into an accumulator to maintain a constant pressure. The accumulator provides the flow to the motor. Motor speeds and direction, hence elevating speeds and direction, are regulated by controlling the flow of the variable-displacement pump or by a valve controlling the flow through two lines leading to the motor.

Electric power drives behave similarly to hydraulic drives. They can elevate or depress at

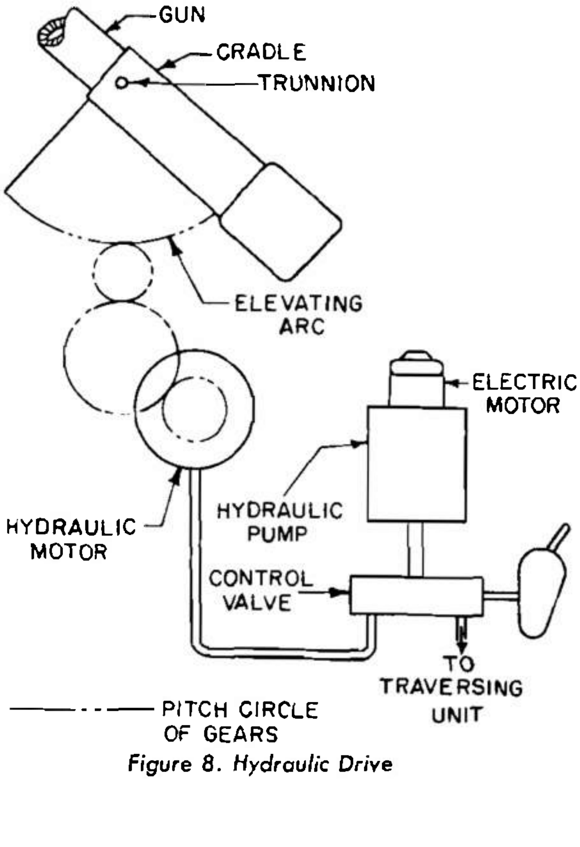


Figure 8. Hydraulic Drive

variable speeds and are readily adaptable to precise automatic control. These characteristics are made available through an amplidyne system whose voltage output to the drive motor can vary according to the elevating speed requirements.

C. DIRECTORS AND OTHER CONTROLS

17. Combining the information of its tracking devices with that received from range finders, directors compute firing data electronically and transmit these data as signals to the guns. The sending and the receiving apparatus are synchronized with the elevating system. In a totally automatic installation, the difference in signals (called the error signal) between those generated in the director and those on the gun correspond to the off-target position of the gun. Responding to the signal, the power drives elevate the gun until the error disappears, indicating on-target position. In installations not totally automatic, the gunner controls the elevating operation. He is guided by a pointer on a dial which indicates the firing position as determined by the director. A second pointer on the dial, synchronized with the elevating mechanism, represents the actual gun position. To bring the gun into firing position, the gunner merely adjusts the weapon until the second pointer matches the first thus indicating an elevation which corresponds with the firing data.

There are several methods of controlling power elevation without the aid of directors. These are are basically the same, differing only in the method of manipulation. One method has control by handwheel with the drive operating as a power assist. Another has handles turning on a vertical axis similar to a steering wheel. Still another has a "joy stick" arrangement. Finally, control may be achieved by manipulating a ball which serves as the initiating component of the control unit.

18. With the help of Figure 9, the mechanics of a ball control unit will be discussed in some detail to illustrate a typical unit. Aside from the ball and gear trains, the controller has seven basic components. These are:

A. Magnetic clutch, direct engagement between ball and *G*.

B. Centering cam, renders *G* inoperative when clutch is disengaged.

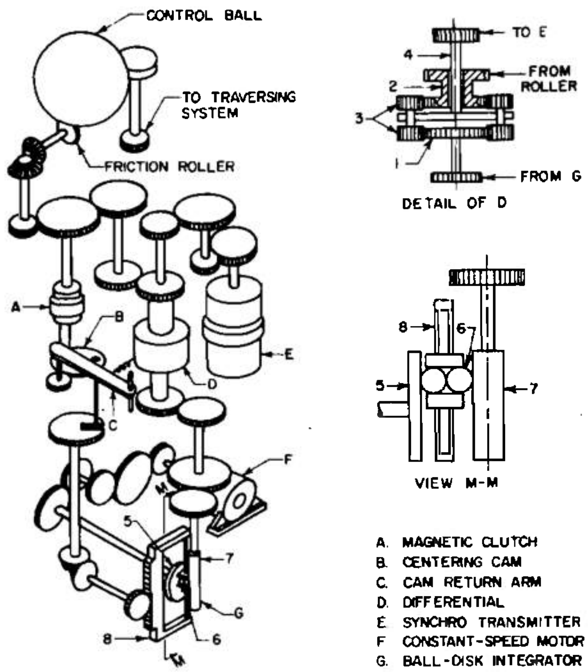


Figure 9. Ball Drive Controller (Schematic of Elevating Unit)

C. Cam return arm, returns cam to neutral position.

D. Differential, provides access of gear train to *E*.

E. Synchro transmitter, sends operating signals to power drive.

F. Constant speed motor, provides continuous motion of *G*.

G. Ball-disk integrator, regulates elevating speed.

Direct manual control is exercised by disengaging the motor driven unit and directing all motion to the synchro transmitter by the ball. The sequential operation follows:

a. Disengage Clutch *A*.
b. Control Arm (*C*) centers Cam (*B*) by spring load.
c. Integrator moves into neutral position with Balls (6) centered on Disk (5).
d. No peripheral motion exists and none is transmitted to gear train linking Integrator (*G*) and Differential (*D*).
e. Gear (1) in differential is thus held stationary and motion from ball is directed through gears (2 and 3) to Synchro Transmitter (*E*).

Both angular velocity and displacement of the tipping parts are proportional respectively to the corresponding activity of the control ball.

19. Assisted manual control becomes available when the clutch is engaged. The gear train between control ball and integrator is uninterrupted. The control ball now exercises control of the elevating direction and speed merely by its own displacement which regulates the displacement of the yoke (8) of the integrator. As the yoke moves in either direction from the axis of the constant speed disk (5), it carries the balls (6) toward faster peripheral motion. The balls transmit this motion to the roller (7) and through the gear train to (1) of the differential and eventually to the synchro transmitter. Gear (2) of the differential is held stationary by the motionless control ball. Depression is achieved by moving the balls of the integrator across the disk axis. Action is stopped by reversing the control ball until the cam centers in its neutral position or by releasing the clutch.

D. STABILIZERS

20. The pitching and slewing of a moving tank disturbs its accuracy by creating an unstable firing base which, if uncompensated, might compel the vehicle to come to a halt for firing. During combat this maneuver renders the tank prohibitively vulnerable. Compensations for erratic motion are therefore provided by stabilizers whose primary function is to maintain the pointing of the gun on its target. Gyroscopes are well suited for this purpose. To complete the stabilizer system, some device, either hydraulic, electric or other type, must be provided to convert the reactions of the gyroscope into signals and transmit them to the elevating system which then responds accordingly and corrects elevation misalignment between gun and target. Stabilized systems need not differ basically from the conventional systems except for (1) inclusion of a gyro in the feed back circuit for each of the two planes of control, and (2) use of a control circuit that will exploit the momentum of the rotating parts to conserve power.

IV. DESIGN REQUIREMENTS

A. GENERAL DESIGN DATA

21. Four basic design requirements are essential to any elevating mechanism, namely, control, power transmission, precision, and sensitivity. Control is exercised by two mediums; the fire control equipment which aims the weapon but is not considered part of the mechanism, and the limit stops and locking device which maintain the aim with respect to the elevation. The locking device, considered part of the elevating mechanism, may be a brake, a clutch, or an irreversible screw. Limit stops, usually hydraulic buffers, prevent overtravel. Power, whether manual or mechanical, is generally transmitted by a gear train. Precision depends on the quality of manufacture, particularly in its relation with backlash while sensitivity involves the gear ratio. These requirements, plus the location of all components, become the basis of elevating mechanism design.

22. The gear train, no matter how simple, is the principal part of the elevating mechanism. If the train is reversible, a brake or other type locking device is needed to preclude reverse motion. The gear ratio should be considered first. Not only does the ratio prescribe the effort at the source, it also affects the sensitivity. A high gear ratio means slow elevation with respect to applied motion; a low ratio means correspondingly faster action. Although a high gear ratio requires only a low torque output at the power source, the power requirements to meet identical performance specifications are unaffected. For example, assume that the torque at the elevating arc, incorporating the efficiency of the gear train, is 2240 lb-ft. For slow elevation, a gear ratio of 640:1 reduces it to 42 lb-in at the handwheel; a ratio of 320:1 reduces it to 84 lb-in. For fast elevation, say 33.7 degrees per second, the shaft output requires 2 horsepower whether the motor turns at 3600 rpm through a gear reduction of 640:1 or at 1800 rpm through a reduction of 320:1. Specified handwheel effort and motor characteristics with sensitivity requirements will establish the gear ratio. Thereafter, type, number, size, and location of the gear train components may be determined.

In addition to the available data, the design of the gear train involves some iterative procedure, more so if mechanically powered since preliminary estimates of both efficiency and inertia must be made. Only the efficiency need be estimated for the manually operated train since accelerating forces are low enough to be negligible. Final computed data must agree reasonably well with the estimates before the preliminary concepts are acceptable. In short, to assure an adequate power supply, the effort required to elevate the tipping parts, including the moving components of the elevating mechanism, should be safely under that available at the power source.

23. The design data is computed from statics and dynamics of the moving masses. The data on statics include the weight moment after equilibration and the torque due to frictional resistance of the trunnion bearings. The unbalanced equilibrator moment is

$$T_e = M_w - M_e \qquad (1)$$

where M_e = equilibrator moment
M_w = weight moment.

The static frictional torque in the trunnion bearings is

$$T_b = \mu F_T r_b \qquad (2)$$

where F_T = load on both trunnions
r_b = bearing radius
μ = coefficient of friction

When recoil forces are present, F_T* increases appreciably thereby increasing the torque, T_b. The additional torque must be considered if the weapon fires while elevating.

24. Another component of the torque at the elevating arc, the firing couple, is generated during firing and is due to the eccentricity of the applied and inertia loads about the trunnion axis. According to Figure 7, the component of the firing couple affecting elevation is

$$T_f = aF_g - bF_a \qquad (3)$$

where F_a = inertia force of recoiling parts
F_g = propellant gas force

Careful design should hold this torque to a minimum, the object being to have the trunnion axis, and the mass center of the recoiling parts, lie on the bore center. This alignment is not always possible. Space limitations and required structural locations may cause an unequal distribution of weight and create an unbalance about the bore axis. And, regardless of the care exercised, manufacturing tolerances may augment this unbalance. If the firing torque is transmitted through the gear train, the effect on the power supply may be significant. If locked out, only the gears between the elevating arc and the locking device will be disturbed.

*The method for computing the bearing load is presented in Chapter VIII of Reference 3.

25. This couple can be modified to a degree by shifting the trunnion axis. No location can maintain a zero torque throughout the entire firing cycle. This is especially true during the propellant gas period as the gas force varies from a maximum to zero. For example, consider a hypothetical gun whose recoil force is K and the propellant gas force ranges from $16K$ to 0, thus

$$F_a = F_g - K \qquad (3a)$$

Using the geometry of Figure 7 when $a=0$, the firing couple varies from $T_f = -15Kd$ to $T_f = Kd$. When zero torque is desired at maximum gas pressure,

$$a = 15d \qquad (3b)$$

$$b = a + d = 16d \qquad (3c)$$

The firing torque as F_g reduces to zero varies from $T_f = 0$ to $T_f = 16Kd$.

TABLE 1. FIRING TORQUE WITH MAXIMUM DISPLACEMENT OF TRUNNION AXIS

F_g	F_a	aF_g	bF_a	T_f
$16K$	$15K$	$240Kd$	$240Kd$	0
$15K$	$14K$	$225Kd$	$224Kd$	Kd
$14K$	$13K$	$210Kd$	$208Kd$	$2Kd$
—	—	—	—	—
—	—	—	—	—
$2K$	K	$30Kd$	$16Kd$	$14Kd$
K	0	$15Kd$	0	$15Kd$
0	$-K$	0	$-16Kd$	$16Kd$

Now assume an intermediate value, say $a=7d$ and $b=8d$.

TABLE 2. FIRING TORQUE WITH INTERMEDIATE DISPLACEMENT OF TRUNNION AXIS

F_g	F_a	aF_g	bF_a	T_f
$16K$	$15K$	$112Kd$	$120Kd$	$-8Kd$
$15K$	$14K$	$105Kd$	$112Kd$	$-7Kd$
$14K$	$13K$	$98Kd$	$104Kd$	$-6Kd$
—	—	—	—	—
—	—	—	—	—
$9K$	$8K$	$63Kd$	$64Kd$	$-Kd$
$8K$	$7K$	$56Kd$	$56Kd$	0
$7K$	$6K$	$49Kd$	$48Kd$	Kd
—	—	—	—	—
—	—	—	—	—
$2K$	K	$14Kd$	$8Kd$	$6Kd$
K	0	$7Kd$	0	$7Kd$
0	$-K$	0	$-8Kd$	$8Kd$

These examples show a varying firing torque during the propellant gas period no matter where the trunnion axis is located. After this period only the inertia forces induced by the recoil mechanism are acting and for a considerably longer time than the gas period. The obvious solution is to locate the trunnion axis in the plane of the recoiling mass center. Except for the gas period, the firing torque is now always zero. But, as mentioned previously, this position may not be readily available or easily located while in the design state. Under such conditions, it is advisable to locate the trunnion axis in the plane of the bore center inasmuch as the large inertia forces of the propellant gas period and hence, maximum torque, appear only briefly while the sustained load and corresponding torque during recoil are considerably smaller.

26. The fourth component of the torque at the elevating arc is that required to accelerate the tipping parts.

$$T_a = \Phi \alpha \tag{4}$$

where Φ = mass moment of inertia of the tipping parts about the trunnions

α = maximum elevating acceleration

The general expression for maximum torque at the elevating arc is

$$T_E = T_a + T_b + T_f + T_e \tag{5a}$$

Note that for double recoil weapons, an additional torque is induced by the secondary recoil acceleration.* The maximum torque in this case will be

$$T_E = T_a + T_b + T_f + T_e + T_t \tag{5b}$$

According to Figure 7, the torque induced by secondary recoil acceleration

$$T_t = F_t \, r_t \sin \phi \tag{5c}$$

where F_t = inertia force of tipping parts due to secondary acceleration

r_t = radius from trunnions to CG of tipping parts

ϕ = angle which r_t makes with the horizontal

Recapitulating, with emphasis on the various conditions: For a level weapon elevated manually and not firing, $T_a \approx 0$ since accelerations are negligible and $T_f = 0$. Therefore

$$T_E = T_b + T_e \tag{6}$$

For a weapon elevated by mechanical or electric power but not firing

* Reference 4, Chapter IV.

$$T_E = T_a + T_b + T_e \tag{7}$$

B. GEAR TRAIN

27. The gear train, as a mechanical transformer, changes the large elevating gear torque to a lesser value at the power source. For a mechanism operated by handwheel, the ratio of the two values is the gear ratio modified by the efficiency. Refer to Figure 10. The gears have even numbers, the

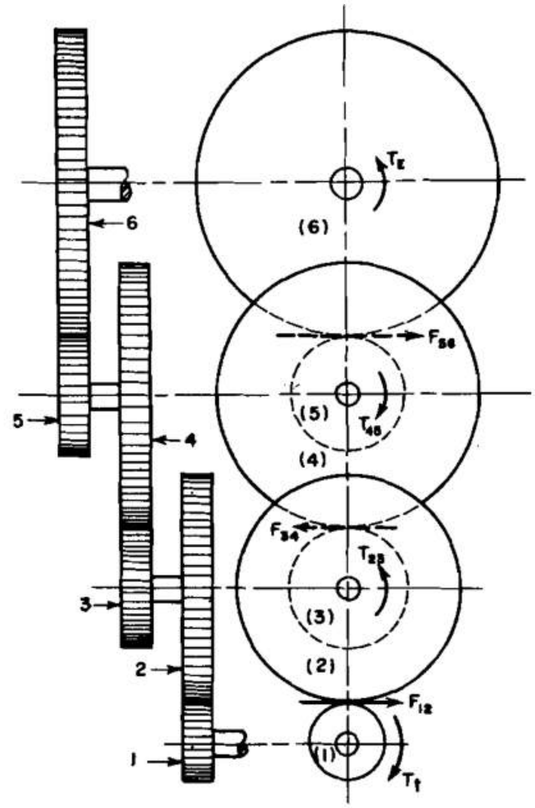


Figure 10. Loading Diagram of Elevating Gear Train

pinions odd. Beginning with the elevating arc (gear) or No. 6, trace the torque through the system by converting it to gear tooth load and back again to torque, taking the efficiency, η_g, into account at each mesh. Thus the gear tooth load between pinion 5 and gear 6 is

$$F_{56} \frac{T_E}{R_{p6}} \tag{8}$$

The torque in gear 4 and pinion 5 becomes

$$T_{45} = \frac{1}{\eta_g} R_{p5} F_{56} = \frac{1}{\eta_g} T_E \frac{R_{p5}}{R_{p6}} \tag{9}$$

The load between pinion 3 and gear 4 is

$$F_{34} = \frac{T_{45}}{R_{p4}} = \frac{1}{\eta_g} T_E \frac{R_{p5}}{R_{p4} R_p} \qquad (10)$$

The torque in gear 2 and pinion 3 becomes

$$T_{23} = \frac{1}{\eta_g} R_{p3} F_{34} = \frac{1}{\eta_g^2} T_E \frac{R_{p3} R_{p5}}{R_{p4} R_{p6}} \qquad (11)$$

The load between pinion 1 and gear 2 is

$$F_{12} = \frac{T_{23}}{R_{p2}} = \frac{1}{\eta_g^2} T_E \frac{R_{p3} R_{p5}}{R_{p2} R_{p4} R_{p6}} \qquad (12)$$

And, the torque at pinion 1 or at the power source becomes

$$T_m = \frac{1}{\eta_g} R_{p1} F_{12} = \frac{1}{\eta_g^3} T_E \frac{R_{p1} R_{p3} R_{p5}}{R_{p2} R_{p4} R_{p6}} \qquad (13)$$

This is the torque required at the power source to turn the parts. Expressed generally

$$T_m = \frac{1}{\eta_g^n} \frac{T_E}{r_g} \qquad (14)$$

where n = number of gear meshes
r_g = gear train ratio
η_g = 0.98 to 0.99* efficiency of spur or bevel gears

When a pinion is replaced by a worm whose efficiency is η_w

$$T_m = \frac{1}{\eta_w \eta_g^{n-1}} \frac{T_E}{r_g} \qquad (15)$$

For the worm being the driving member

$$\eta_w = \frac{\cos\beta - \mu\tan\lambda\dagger}{\cos\beta + \mu\cot\lambda} \qquad (16)$$

where β = pressure angle
λ = lead angle
μ = coefficient of friction

When frictional losses in the thrust bearing of the worm are considered, the efficiency

where D_b = effective diameter of the thrust bearing
D_w = pitch diameter of worm
μ_b = coefficient of friction of the thrust bearing

Expressed in terms of the composite efficiency of the gear train

$$T_m = \frac{1}{\eta} \frac{T_E}{r_g} \qquad (18)$$

28. The inertia of the gear train contributes to the total required effort for motorized elevating mechanisms. As it is being accelerated, the torque on each gear progresses through the train similarly to the one coming from the elevating arc. Referring to Figure 10, start at the integral gear No. 45 and express all accelerations in terms of the elevating acceleration.

$$T_\alpha = \frac{1}{\eta_g^{n-1}} T'_{45} \frac{R_{p1} R_{p3}}{R_{p2} R_{p4}} + \frac{1}{\eta_g^{n-2}} T'_{23} \frac{R_{p1}}{R_{p2}} + \frac{1}{\eta_g^{n-3}} T'_1 \qquad (19)$$

but

$$T'_{45} = \Phi_{45}\alpha_{45},\ T'_{23} = \Phi_{23}\alpha_{23},\ T'_1 = \Phi_1\alpha_1$$

and

$$\alpha_{45} = \alpha \frac{R_{p6}}{R_{p5}},\ \alpha_{23} = \alpha \frac{R_{p4} R_{p6}}{R_{p3} R_{p5}},\ \alpha_1 = \alpha \frac{R_{p2} R_{p4} R_{p6}}{R_{p1} R_{p3} R_{p5}}$$

therefore, through substitution with $n=3$

$$T_\alpha = \left(\frac{1}{\eta_g^2} \Phi_{45} \frac{R_{p1} R_{p3} R_{p6}}{R_{p2} R_{p4} R_{p5}} + \frac{1}{\eta_g} \Phi_{23} \frac{R_{p1} R_{p4} R_{p6}}{R_{p2} R_{p3} R_{p5}} + \Phi_1 \frac{R_{p2} R_{p4} R_{p6}}{R_{p1} R_{p3} R_{p5}}\right) \alpha \qquad (20)$$

A conservative estimate of this torque assumes the gear train efficiency for all components rather than the summation of the effects of the individual efficiencies for each gear unit. The resulting error is too small to be significant since the inertia of the gear train will be far less than that of the tipping parts and in practically all cases may be ignored.

$$\eta_w = \frac{\cos\beta - \mu\tan\lambda}{\cos\beta\left(1 + \mu_b \frac{D_b}{D_w}\cot\lambda\right) + \mu\cot\lambda\left(1 - \mu_b \frac{D_b}{D_w}\tan\lambda\right)} \ddagger \qquad (17)$$

* Reference 5, pages 320 and 346.
† Reference 5, page 355.
‡ Reference 6, page 389.

When desired, torque necessary to drive the gear train is

$$T_a = \frac{\alpha}{\eta_g^{n-1}}\left(\Phi_{45}\frac{R_{p1}\,R_{p3}\,R_{p6}}{R_{p2}\,R_{p4}\,R_{p5}} + \Phi_{23}\frac{R_{p1}\,R_{p4}\,R_{p6}}{R_{p2}\,R_{p3}\,R_{p5}} + \Phi_1\frac{R_{p2}\,R_{p4}\,R_{p6}}{R_{p1}\,R_{p3}\,R_{p5}}\right) \tag{21}$$

and the required torque output at the shaft of the power source is

$$T_m = \frac{T_E}{\eta_g^{\,n}\,r_g} + T_a \tag{22}$$

29. The choice of gear train ratio depends on the type of elevation on one end and the type of power supply on the other. For manual operation where speeds are not critical, a ratio is chosen which demands a reasonable physical effort. Coupled with this requirement, a 640:1 gear ratio proves advantageous as it represents a 10- mil-per-turn response to give the gunner a rough but convenient counter. For power transmission, speed is a major criterion, the gear ratio becoming the ratio between motor rpm and the elevating speed. Motor choice should be limited to those having conventional speeds which allows some flexibility in the selection of the gear ratio but restricts motor selection to the standard speeds in this range. Elevating speed depends on weapon type. An angular velocity of 10 degrees per second is sufficient for field artillery since targets are usually stationary. For tanks or antiaircraft guns, elevation may be as fast as 55 degrees per second. Quick response is also necessary for the latter type, thus requiring high-performance servosystems and therefore a more elaborate design approach. For these units, the time rate of power increases as limited by the inertia of the elevating motor and its gear train is recognized as the design approach to the servosystem. The power source must be capable of developing peak torque at peak acceleration while the mount is elevating at peak speed. Investigations have shown that optimum power requirements and speed reductions can be determined to fit the particular need of an elevating system. Some of these relationships are expressed in Equations 23, 24, and 25.*

$$r_g = \Upsilon r_n \tag{23}$$

$$\Upsilon = \frac{1}{2}\frac{t_t}{t_m} \tag{24}$$

$$\Upsilon P_m = 2P_t \tag{25}$$

where P_m = power generated by motor

P_t = power required to rotate the tipping parts

$r_n = \dfrac{\text{peak speed rating of motor}}{\text{peak rotational speed of mount}}$

r_g = required gear ratio

t_m = time constant of motor

t_t = time constant of the tipping parts

$\Upsilon = \dfrac{\text{maximum operating motor speed}}{\text{peak speed rating of motor}}$

The time constant of the motor is defined as the time required to bring the motor, running free of load, from standstill to maximum operating speed at maximum angular acceleration. The time constant of the tipping parts is defined as the time required to bring the tipping parts to maximum speed from a standstill at maximum elevating acceleration. The above expressions will serve as a guide for the elevating mechanism designer who, although not necessarily the power drive designer, must compile the essential design data, meanwhile keeping in touch with the power drive specialist to make sure that these data are not too demanding. The inertia of the rotating components of the motor is a major design parameter in a quick-response elevating mechanism. This inertia must be considered with that of other moving parts when designing or selecting a motor to accelerate all moving components of the entire elevating system.

30. The maximum elevating acceleration depends on the tactical use of the weapon. For instance, field artillery fires on fixed targets, therefore requiring minutes rather than fractions of a second for aiming, whereas guns firing on moving targets must have high acceleration to swing quickly toward them. Thus, tank and antiaircraft guns must have angular acceleration on the order of 0.5 rad/sec^2. Tanks while maneuvering pitch at angular acceleration far in excess of the elevating acceleration of 0.5 rad/sec^2. To preclude inadvertent rotation of the tipping parts the elevating mechanism must serve as a travel lock. Thus the gear train will transmit the torque from tipping parts to turret. This torque is limited to that induced by a maximum acceleration of 55 rad/sec^2. Should pitching accelerations exceed this value, a clutch, somewhere in the gear train, slips at the

* Reference 7.

corresponding torque and permits relative motion between tipping parts and tank. If a gun must be held on target despite pitching of the tank, the inertia of the tipping parts will hold the elevation prescribed by the target. Under this circumstance, the motor need accelerate only the gear train to take up the slack. The required acceleration, equivalent to the pitching acceleration, must be increased by the normal elevating acceleration of 0.5 rad/sec^2 when the required gun movement is opposite to the pitching direction.

If the tipping parts are locked by the elevating mechanism, the high pitching acceleration will excite vibrations in the elastic system composed essentially of gun tube and elevating mechanism components. It is highly probable that the various bending nodes will couple with the oscillating node of the tube as a pendulum suspended by the elevating gear train which is equivalent to a torsion spring. The frictional torque during the oscillation is reversing rapidly and is additive during part of the cycle but is of such a small numerical value that it may be ignored. Far more important is the dynamic response torque which may be sizable, depending on the shape of the ramp of the pitching acceleration-time curve leading up to 55 rad/sec^2. Depending on the terrain over which the tank is traveling, these dynamic augments may exceed the design acceleration. When this happens, the tank structure may absorb a considerable portion of the shock but enough may still reach the gun to cause trouble. In this event, some form of damping device, a relief valve in a hydraulic unit or a slip clutch in a gear train, should be installed to control the excessive dynamic activity.

Assuming that the gear ratio has been established, the type, number, size, and location of the gears are determined next. The relative positions of the elevating arc and handwheel or power drive, in addition to space availability, locate the gears. Accessibility for maintenance also influences the location. The gear train should be protected by a gear box or other suitable enclosure with provision for adequate lubrication. For parallel shafting, spur, helical, or herringbone gears are available. The helical and herringbone are smooth running at high speeds and are comparatively low stressed with the herringbone having the added advantage of neutralizing the induced end thrust of the helical gear. Bevel and worm gears are used for nonparallel shafting, the latter being particularly adapted for high speed or torque ratios. Gears should be few in number and small to minimize inertial and frictional losses. A discussion of "gear trains of least inertia" is available in Reference 6. They must mesh with a minimum of clearance, the total backlash in the train being limited to about one mil. Backlash is sometimes eliminated with an antibacklash device. Tooth loads are important in design but other factors such as speed and accuracy of tooth form must also be considered. However, detailed design procedures are available in most good machine design texts. Also, specialists are always within reach at gear plants for consultation.

C: HANDWHEELS

31. Manual operation of an elevating mechanism begins with a crank or handwheel. Two-handed effort is conducive to smoother and more uniform operation and should be planned for, provided that one hand is not needed to manipulate other controls simultaneously. The smoother action will be less tiring. For two-handed operation, a double crank (see Figure 1) or a double handwheel (Figure 11) serves the purpose. Handwheels, other sighting controls, and associated indicators must be located near each other to offer ready access and easy observation to the gunner. All manipulating equipment such as wheels, handles, levers and push buttons must be large enough to be operated easily with gloved hands during cold weather operations.

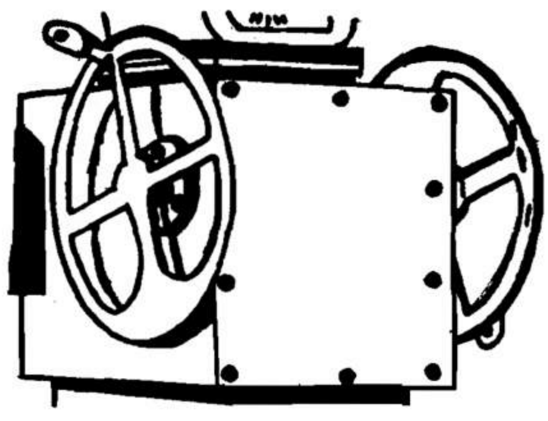


Figure 11. Double Handwheel Mechanism

Handwheels are designed on the basis of the physical effort required to overcome static loads and frictional resistance of the elevating system. Dynamics need not be considered as speeds are necessarily low. Handwheels are normally circular with a handle and a counterweight attached opposite to each other on the rim. Any structural material may be used including cast iron, steel, aluminum, magnesium. Some are cut from plate stock, others are cast. Some work has been done to determine a man's capacity for turning a handwheel. His best effort for sustained operation was achieved while producing 50 lb-in of torque at a turning radius of 7 inches.* For equipment that cannot meet these limits, handwheel force should not exceed 15 pounds. Where elevating loads are extreme, the mechanical advantage of the gear train must be increased to reach the required handwheel effort. These same conditions prevail for handwheels on mechanically powered systems.

D. LOCKING DEVICES

32. A locking device performs two functions. First, as a control measure, it holds the tipping parts in a fixed position during firing by resisting any unbalanced couples which exceed the equilibrator moment and tend to elevate or depress the tipping parts. Second, as a safety measure, it prevents these same couples from reversing through the gear train to spin the handwheel inadvertently and endanger personnel. Worm gearing, brakes, and irreversible clutches are positive locking devices capable of stopping all gear train motion; the brake in either direction, the worm and irreversible clutch in one direction only. On the other hand, friction clutches limit the capacity of gear trains. The slip clutch cannot transmit torque beyond a predetermined limit. Brakes or worm gearing are generally used on stationary mounts where the largest reverse moment is the firing couple. Slip clutches are installed on self-propelled weapons such as tanks to limit the elevating mechanisms to reverse torques much lower than those induced by the maneuvering vehicle. Irreversible clutches of limited capacity may be used as safety devices on any elevating mechanism equipped with a handwheel. If brakes or clutches are installed near the source of the reverse torque, the advantage of lower gear loads obtained here is offset by the large and costly device required to transmit a large operating torque. On the other hand, if near the low-torque, high-speed end of the gear train, a much smaller and less costly one is needed to provide the required low resistance. The advantage gained here more than compensates for the somewhat larger capacity gears required to carry the additional inertial loads.

* Reference 8.

1. Worm Gear

33. The simplest locking device may be a component of the gear train, namely, the irreversible worm. Because of its inherent strength and large mechanical advantage, worm gearing should be near the elevating arc in the gear train sequence to prevent excessive loads from acting on gears that can then be designed for lighter loads.

2. Brakes

34. A gear train with no irreversible feature must rely on brakes to hold the tipping parts motionless. Mechanical, electric, or electrohydraulic brakes may be used. The last two types, when equipped with interlocks, add safety to gun handling. For instance, limit switches are so located that the brakes are applied just before the elevating gun reaches obstructions in the firing path. Brakes also hold the gun in position while it is being loaded. The brake may be placed anywhere in the gear train, its capacity being governed by its proximity to the elevating arc.

3. Clutches

35. Clutches are used either to link power source to gear train or control the amount of torque transmitted. Positive types such as square jaw clutches are recommended for systems which are stopped during clutch operation. This type is small for the power it is capable of transmitting and construction costs are low. Friction types, including cone, disk, and ring clutches, are recommended for systems in motion so that torque may be applied gradually thus reducing the possibility of sudden loading. Slip clutches, a type of friction clutch, transmitting only a limited torque, are used in No-Bak installations and in the elevating systems of tanks to protect the elevating mechanism from large reverse torques. Much more sensitive than brakes or other clutches, the slip clutch relies upon a fairly constant frictional

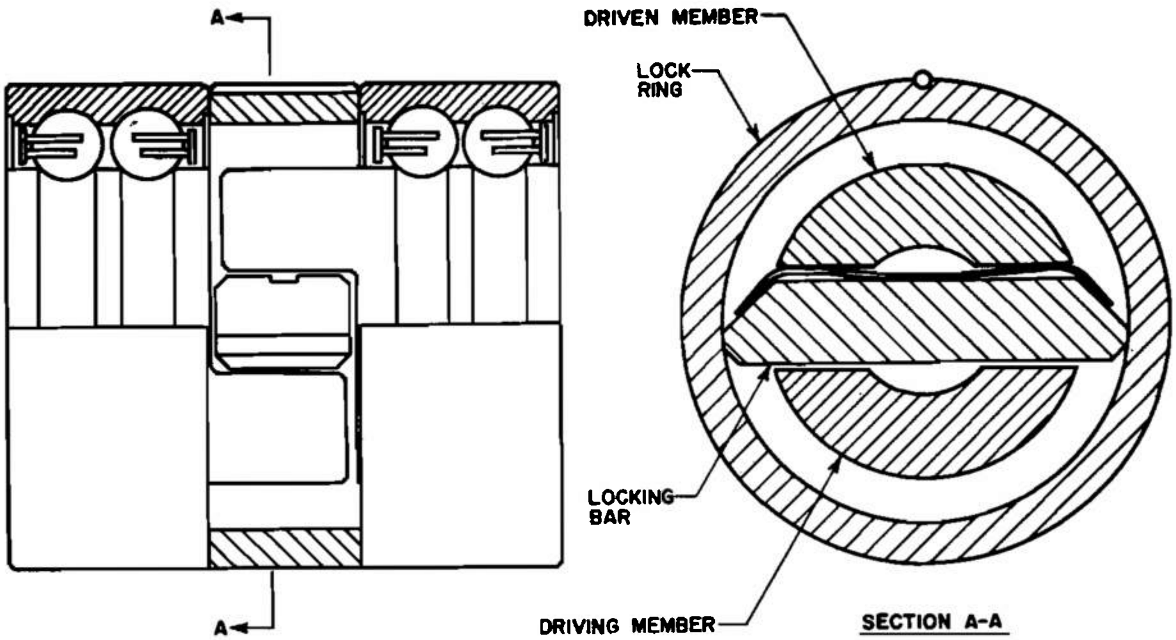


Figure 12. Smith No-Bak Device

resistance. Since the coefficient of friction of its rubbing surfaces is influenced by atmospheric conditions, its performance may become erratic, therefore the undesirable characteristic must be included as a design consideration.

4. Irreversible Clutches

36. Power driven mounts require reversibility but this reversibility should not be extended to the handwheel whose inadvertent spinning may prove dangerous. A special irreversible clutch called a Smith No-Bak Device transmits torque in either direction from handwheel to gear train but precludes a reversal of this activity. Figure 12 illustrates the device. Drive and driven shaft have overlapping ends, a segment extending past the diameter being cut from each. The void thus formed makes room for the spring-loaded locking bar, the shear member transmitting the handwheel effort. Shafts and bar rotate inside the lock ring which is keyed to the housing. The locking bar with ends slightly off center, bears against the inner surface of the ring on the drive shaft side of the diameter. Held firmly between the shafts by a flat spring, it rotates with them in the direction of the applied torque. Torque from the handwheel moves the locking bar toward the center so that its longest dimension lies along the diameter, thus releasing its contact with the lock ring and permitting both shafts to turn. Torque from the other end moves the bar off center and jams it against the lock ring, stopping all motion. Since its capacity is limited, a slip clutch in the gear train, between No-Bak and tipping parts, yields to any excessive torque to relieve the load on the No-Bak and eliminate it as a lock for the whole elevating system.

The recommended approach for the design of brakes and clutches parallels that for gears. Good texts which contain detailed design procedures are readily available. One should be thoroughly familiar with their procedures in order to select suitable components for the initial design concepts. Art as well as science is involved, particularly for slip clutches, therefore, if the elevating mechanism designer is not an expert in this field, one should be consulted before designs are finalized.

E. BUFFERS

37. The elevating range, because of structural interferences and operational difficulties, is limited to an angle slightly below the horizontal to one slightly below the vertical. Because of tactical use and construction features this range may be considerably smaller. When the assigned limits are reached, some arrangement must be available

to stop the tipping parts. On power driven units, limit switches give the signal for torque reversal. Handwheel operation can be stopped at will. But neither provides a positive stop. On small, manually elevated mounts where inertias are low, a filled-in tooth at each end of the elevating arc or mating lugs on tipping parts and top carriage serve as solid stops. On large or power driven mounts where inertias are high, some shock absorbing device must be used to cushion the impact and absorb the energy. Springs are inadequate because of their ability to store energy and bounce back. Hydraulic buffers are ideal because of their energy absorbing characteristics. These buffers are not completely hydraulic since a light spring is needed to return and hold the moving parts in their stand-by position.

Buffers bring the moving parts to a firm but smooth stop and should be designed for the severest condition which means constant deceleration when operating torque and kinetic energy of tipping parts are maximum. Although limit switches stop the power supply and may even reverse the torque at the power source, the buffers should, in the event of signal failure, be capable of absorbing the kinetic energy of the tipping parts with the driving torque fully applied.

38. Restriction of oil flow through a variable orifice develops the resisting force. One method for varying the orifice is shown in Figure 13. Here two grooves, located 180 degrees apart for balance, provide the necessary restriction. Orifice areas, largest at the beginning of the stroke where velocities are highest, gradually grow smaller along the stroke as velocities decrease and eventually become zero. This area at any position along the stroke is

$$a_o = \frac{v_b}{c_o}\left(\frac{\rho A_b^3}{2 F_b}\right)^{1/2} * \qquad (26)$$

where A_b = effective area of buffer piston
c_o = 0.60,† orifice coefficient, for large approach area to discharge area ratios
F_b = buffer force
v_b = buffer velocity at any position, x, of stroke
ρ = mass density of hydraulic fluid

* This equation was developed in Reference 9 and appears there as Equation 36.
† Reference 10, page 261.

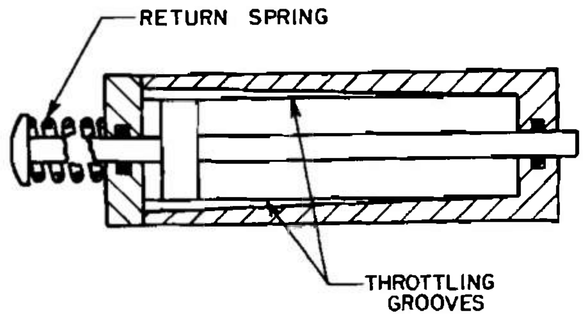
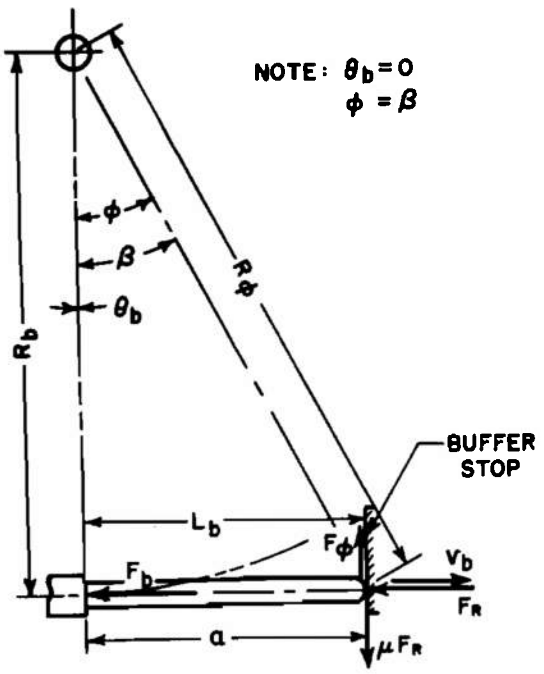


Figure 13. Buffer

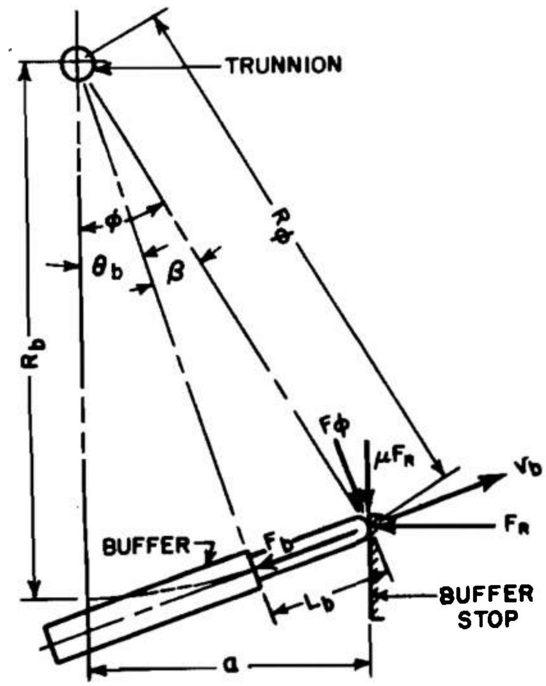
If either the contact velocity or the applied torque, or both, are less than maximum when the buffer is contacted, the force, F_b, will be less but the buffer will still have the capacity to bring the tipping parts to a stop without exceeding the design force. Once the orifice has been established, all subsequent buffing dynamics are fixed. Therefore, when the dynamics of the tipping parts change, v_b and F_b can be found only by long, unwieldy, iterative computation.

39. Buffers may be fixed to the top carriage with their axes tangent to a circular arc passing through the initial point of contact. The mechanics of this type of arrangement are presented in detail in Reference 1. Another type of installation may have the buffers attached to and moving with the tipping parts. Both types are designed on the basis of constant angular deceleration. Actually, the angular displacement of the tipping parts during buffing is so small that geometric proportions vary only slightly within this range thereby inducing a buffer force that is practically constant. Figure 14 is a diagram of the buffer geometry and activity. The geometry is exaggerated for clarity and is based on the location of the buffer and buffer stop with respect to the trunnions and the length of buffer stroke. The definitions of the dimensions in Figure 14 follow.

a = offset distance from trunnion to buffer stop
F_b = buffer force
F_R = reaction of buffer stop
F_ϕ = normal load on buffer rod end
L_b = effective length of buffer rod
R_b = radius from trunnions, perpendicular to line of action



(a) INITIAL CONTACT POSITION



(b) PARTIAL STROKE POSTION

Figure 14. Dynamics of Buffer (Schematic)

R_ϕ = radius to point of contact on buffer stop (variable)
v_b = buffer velocity
θ_b = angular displacement of buffer
β = angular distance between buffer and buffer stop

40. Since F_R is normal to its surface, the buffer stop cannot provide a parallel reaction any larger than the induced frictional resistance of μF_R which opposes the motion of the buffer rod end on the stop. Expressed in terms of the reactions, the buffer force and the normal load on the rod end becomes

$$F_b = F_R \cos \theta_b + \mu F_R \sin \theta_b \qquad (27)$$

$$F_\phi = F_R \sin \theta_b - \mu F_R \cos \theta_b \qquad (28)$$

The torque about the trunnions induced by the buffer forces is

$$T = R_b F_b - L_b F_\phi \qquad (29)$$

But, according to the geometry

$$L_b = R_b \tan \beta \qquad (30a)$$

and

$$\tan \beta = \frac{a - R_b \sin \theta_b}{R_b \cos \theta_b} \qquad (30b)$$

Therefore

$$T = R_b F_b - \frac{a - R_b \sin \theta_b}{\cos \theta_b} F_\phi \qquad (31a)$$

Substituting the expression for F_ϕ in Equation 29 and then for F_R as determined from Equation 28, the torque can now be expressed as

$$T = F_b \left(R_b - \frac{\sin \theta_b - \mu \cos \theta_b}{\cos \theta_b + \mu \sin \theta_b} \; \frac{a - R_b \sin \theta_b}{\cos \theta_b} \right) \qquad (31b)$$

Practical applications will always have small angular displacements during buffing, hence, with θ_b zero just at buffer contact, $\sin \theta_b$ will be small and $\cos \theta_b$ near unity. If the coefficient of friction is appreciable, $\mu \cos \theta_b$ will have considerably more influence in Equation 31b than the almost negligible value of $\mu \sin \theta_b$. For a given torque, as μ increases, F_b decreases, therefore, the conservative design approach should assume μ to be zero and yield the maximum value of F_b. Equation 31b now becomes

$$T = F_b \left(R_b - \frac{a - R_b \sin \theta_b}{\cos \theta_b} \tan \theta_b \right) \quad (31c)$$

Having decided on the radius, R_b, the total buffer stroke, x_b, and the dimension, a, the buffer force can be determined for any position after the total angular displacement is determined from Equation 32 by solving for θ_b (see Sample Problem H).

$$R_b \sin \theta_b + L_b \cos \theta_b = a \quad (32)$$

where $L_b = a - x_b$

Knowing θ_{bm} at x_b, and ω, the elevating velocity, the constant angular deceleration becomes

$$\alpha_b = \frac{\omega^2}{2\theta_{bm}} \quad (33)$$

The decelerating torque induced by the buffers consists of two components, both being constant. Since α_b is constant, the component derived from the rotational energy of the elevating system is constant.

$$T_{ab} = \Phi_e \alpha_b \quad (34)$$

where the effective mass moment of inertia is

$$\Phi_e = \Phi + \Phi_{67} \left(\frac{R_{p8}}{R_{p7}}\right)^2 + \Phi_{45} \left(\frac{R_{p8}\, R_{p6}}{R_{p7}\, R_{p5}}\right)^2 + \Phi_{23} \left(\frac{R_{p8}\, R_{p6}\, R_{p4}}{R_{p7}\, R_{p5}\, R_{p3}}\right)^2 + \Phi_1 r_g^2 \quad (35)$$

The other component is the torque composed mostly of the torque applied through the gear train by the motor. Since the frictional torque and residual equilibrator torque are still acting, the elevating torque is reduced by these values to the original required accelerating torque. Thus, the second component of the required decelerating torque is merely the accelerating torque, T_a (see Equations 4 and 7). The total torque to be induced by the buffers is

$$T = T_{ab} + T_a \quad (36)$$

Two other parameters needed for buffer design are the velocity at any position and the buffer travel. If ω_o is the initial angular velocity, the buffer velocity at any position is

$$v_b = R_b \sqrt{\omega_o^2 - 2\alpha\theta_b} \quad (37)$$

and the buffer travel

$$x_b = a - L_b \quad (38)$$

V. SAMPLE PROBLEMS

A. HEAVY ARTILLERY WITH POWER DRIVE

41. Determine the characteristics for an elevating mechanism based on the following data:

W_t = 14,000 lb, weight of tipping parts
F_T = 50,000 lb, trunnion load
T_e = 26,800 lb-in, residual weight moment
Φ = 174,000 slug-ft^2, mass moment of inertia of tipping parts about the trunnions
α = 0.10 rad/sec^2, elevating acceleration
ω = 15°/sec, maximum elevating velocity

From Equation 4

$$T_a = \Phi\alpha = 174{,}000 \times 12 \times 0.10 = 209{,}000 \text{ lb-in}$$

From Equation 2,

$$T_b = \mu F_T r_b = 1{,}500 \text{ lb-in}$$

where

r_b = 3.0 in, trunnion bearing radius
μ = 0.01, coefficient of friction

From Equation 7

$$T_E = T_a + T_b + T_e = 237{,}300 \text{ lb-in}$$

This is the required torque at the elevating arc.

42. The gear ratio of the powered elevating mechanism is determined from the required elevating velocity and the rated speed of the motor which is assumed to be 750 rpm.

$$\omega = 15°/\text{sec} = \frac{15}{360} \times 60 = 2.5 \text{ rpm},$$

elevating speed

ω_m = 750 rpm, motor speed

$$r_g = \frac{\omega_m}{\omega} = 300, \text{ gear ratio}$$

The selection of the gear train configuration is a matter of geometric convenience although sizes of pinions and gears should be compatible. Figure 15 is a schematic illustration of the train whose gear ratio is

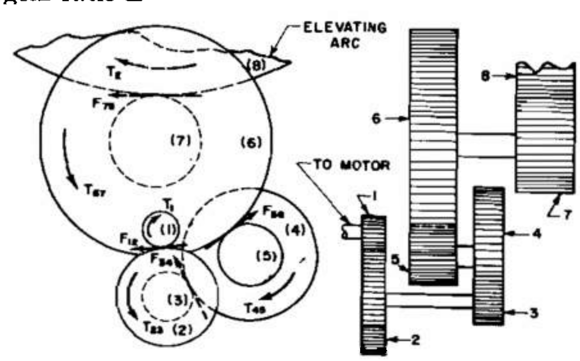


Figure 15. Elevating Gear Train

$$r_g = \frac{R_{p8}}{R_{p7}} \times \frac{R_{p6}}{R_{p5}} \times \frac{R_{p4}}{R_{p3}} \times \frac{R_{p2}}{R_{p1}}$$

$$= \frac{36}{4.5} \times \frac{6.25}{2} \times \frac{4.5}{1.5} \times \frac{4}{1} = 300$$

where R_{px} = pitch radius of the respective gears Table 3 is the accumulated data of the gear train. The procedure for obtaining the data is based on the material on spur gears in Reference 6. These are precision gears and although a more elaborate method is available in the same reference, the one presented here is adequate for illustrating a gear design procedure. Since the teeth in pinions are weaker than the mating gear teeth, the calculations are based on pinion dimensions, No. 5 being selected for the analysis.

TABLE 3. GEAR TRAIN DESIGN DATA, STATIONARY MOUNT

Gear	R_p (in)	T (lb-in)	F_g (lb)	ω (rpm)	v_p (fpm)	c_v	$1000y'$
8	36	237300	6590	2.5	47.1	.918	< .52
7	4.5	30200	6590	20	47.1	.918	.52
6	6.25	30200	4840	20	65.8	.903	.29
5	2	9900	4840	62.5	65.8	.903	2.20
4	4.5	9900	2200	62.5	147.5	.855	.25
3	1.5	3400	2200	187.5	147.5	.855	1.76
2	4	3400	850	187.5	393	.764	.13
1	1	870	850	750	393	.764	1.60

Gear	N	C_F	P_c (in)	F_e (in)	σ (lb/in²)	C_i	F_w (in)
8	432	3.42	.524	1.792	<50000	.628	2-7/8
7	54	3.42	.524	1.792	50000	.628	2-7/8
6	75	3.05	.524	1.597	38800	.85	1-7/8
5	24	3.05	.524	1.597	50000	.85	1-7/8
4	81	3.25	.349	1.135	38500	.90	1-1/4
3	27	3.25	.349	1.135	50000	.90	1-1/4
2	112	3.90	.224	.787	34200	.90	7/8
1	28	3.90	.224	.787	44600	.90	7/8

To assure good tooth contact and a uniform load distribution over the width, the ratio, C_F, of the effective face width to circular pitch should lie between 2.5 and 4.* For preliminary estimates, assume

$$C_F = \frac{F_e}{P_e} = 3.25$$

If the results obtained from this ratio are not reasonable, *e.g.*, the number of teeth is not a whole number, the parameters will be adjusted to make them reasonable. By virtue of their position in the train and by following the procedure indicated by Equation 10, the tooth load where pinion 5 and gear 6 mesh is

$$F_{56} = \frac{T_{67}}{R_{p6}} = \frac{1}{\eta_g} \frac{T_E}{R_{p8}} \times \frac{R_{p7}}{R_{p6}}$$

$$= \frac{1}{.98} \times \frac{237{,}000}{36} \times \frac{4.5}{6.25} = 4840 \text{ lb}$$

The angular velocity of pinion No. 5 is

$$\omega_5 = \omega_6 \frac{R_{p6}}{R_{p5}} = 20 \frac{6.25}{2} = 62.5 \text{ rpm}$$

The linear velocity at the pitch circle is

$$v_{p5} = \frac{2\pi}{12} \omega_5 R_{p5} = \frac{2\pi}{12} \times 62.5 \times 2 = 65.8 \text{ ft/min}$$

According to Equation 13-8 of Reference 6

$$y' = \frac{F_{56}}{\sigma_e c_v C_F D_p^2} = .00206$$

where σ_e = 50,000 lb/in², endurance limit of gear teeth based on a factor of safety of 2.0 and a yield strength of 100,000 lb/in²

C_F = 3.25 ratio, face width to circular pitch*

D_p = 4.0 in, pitch diameter of pinion No. 5

$c_v = 1 - (\sqrt{v_{p5}}/84) = .903$, Barth's velocity factor†

In Figure 16‡, locate the intersection of the horizontal line corresponding to y' = .00206 and the curve for the 20° ASA stub tooth. The intersection falls on the vertical line for N_5 = 25 which yields a diametral pitch of

$$P_d = \frac{N_5}{D_{p5}} = \frac{25}{4} = 6.25$$

This number in itself is reasonable but when the number of teeth in the mating gear are calculated

$$N_6 = P_d D_{p6} = 6.25 \times 12.5 = 78.125$$

a number which is absurd.

If N_5 = 24 teeth (nearest reasonable number), then

$$P_d = \frac{24}{4} = 6$$

and

$$N_6 = 6 \times 12.5 = 75 \text{ a whole number}$$

* Reference 6, page 334

* Reference 6, page 334.
† Reference 6, page 333.
‡ Variations of this chart are readily available in many texts on gear design.

Now returning to Figure 16, for $N_5 = 24$, $y'_5 = .0022$ and for

$$N_6 = 75,\ y'_6 = .00029$$

Solving for the corrected value of C_F from Equation 13-8 of Reference 6

$$C_F = \frac{F_{56}}{\sigma_e\, c_v\, D_{p5}^2\, y'_5} = 3.05$$

$$P_c = \frac{\pi}{P_d} = .524$$

$F_e = P_c\, C_F = 1.597$ in, effective face width

In the second reduction of the gear train, according to Table B-5 of Reference 6, the inbuilt factor is

$$C_i = .85$$

Therefore the actual face width is

$$F_w = \frac{F_e}{C_i} = 1.879 \text{ in or } 1\text{-}7/8 \text{ in}$$

The stress on the gear, which is always lower than on the pinion is

$$\sigma = \frac{F_{56}}{c_v\, C_F\, D_{p6}^2\, y'_6} = 38{,}800 \text{ lb/in}^2$$

43. If the actual tooth width is greater than 2.0 in as in the first reduction at gear 8 and pinion 7, the inbuilt factor is solved from the equation

$$C_i = .667 - .0135\, F_w{}^*$$

but

$$F_w = \frac{F_e}{C_i} = \frac{1.792}{.667 - .0135\, F_w}$$

and

$$F_w^2 - 49.4\, F_w + 132.5 = 0$$

$$F_w = 2.85$$

$$C_i = \frac{F_e}{F_w} = .628$$

The face width, if not originally a nominal size, is increased to the nearest nominal size and the stresses recomputed. For instance, the actual face width of pinion No. 1 is $F_w = .867$ in. The next nominal size is $F_w = 7/8$ in. The face width to circular pitch ratio is

$$C_F = \frac{F_w}{P_c} = \frac{.875}{.224} = 3.90$$

* Reference 6, page 339.

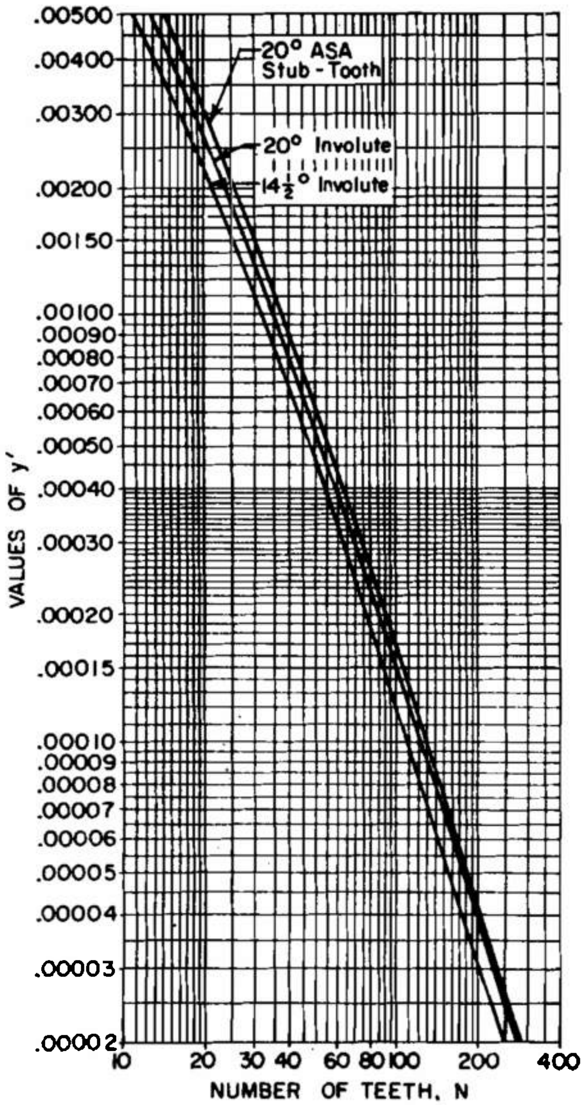


Figure 16. Tooth Form Factor

Correcting the effective width and using value of 0.90 for C_i, obtained in Table B-5 of Reference 6

$$F_{e1} = C_i\, F_w = 0.90 \times .875 = .787 \text{ in}$$

This value appears in Table 3

$$\sigma = \frac{F_g}{c_v\, C_F\, D_{p1}^2\, y'_1}$$

$$\sigma = \frac{850}{.764 \times 3.90 \times 4 \times .0016} = 44{,}600 \text{ lb/in}^2$$

44. The torque at pinion No. 1, the drive motor pinion, is shown in Table 3 as $T_1 = 870$ lb-in. The gears in the train are too small to add significantly to this torque. Later, calculation of a gear train

composed of larger gears will show some difference in the horsepower required.

$HP = T\omega/33000$ when ω is in radians per minute

$$= \frac{870}{12} \times \frac{2\pi\, 750}{33000} = 10.3$$

This is the maximum required horsepower at the motor shaft when the gear train is not included. The added effort to drive the gear train is almost negligible as demonstrated by the calculation below. The mass moments of inertia of the gears are listed in Table 4. Each gear is assumed to be a solid disk whose thickness is the face width and its radius the pitch radius. The assumption is conservative since the gears can be made lighter simply by removing material from the region between rim and hub.

TABLE 4. MASS MOMENTS OF INERTIA OF GEARS, STATIONARY MOUNT

Gear	R_p	R_p^4	F_w	Φ_z	Φ_{zz}
7	4.5	410	2.875	1.37	4.69
6	6.25	1525	1.875	3.32	—
5	2	16	1.875	.035	—
4	4.5	410	1.25	.595	.630
3	1.5	5	1.25	.0073	
2	4	256	.875	.260	.2673
1	1	1	.875	.00102	.00102

where $\Phi_z = \frac{1}{2} M R_p^2 = \frac{\delta\pi}{2g} R_p^4 F_w$

$= .00116\, R_p^4 F_w$ lb-in-sec²

F_w = face width, in

R_p = pitch radius, in

g = 386.4 in/sec²

δ = .285 lb/in³, density of steel

From Equation 21

$$T_a = \frac{\alpha}{\eta_g^{n-1}} \left(\Phi_{67} \frac{R_{p1} R_{p3} R_{p5} R_{p8}}{R_{p2} R_{p4} R_{p6} R_{p7}} + \Phi_{45} \frac{R_{p1} R_{p3} R_{p6} R_{p8}}{R_{p2} R_{p4} R_{p5} R_{p7}} + \Phi_{23} \frac{R_{p1} R_{p4} R_{p6} R_{p8}}{R_{p2} R_{p3} R_{p5} R_{p7}} + \Phi_1 \frac{R_{p2} R_{p4} R_{p6} R_{p8}}{R_{p1} R_{p3} R_{p5} R_{p7}} \right)$$

$$= \frac{0.10}{.98^3} \left(4.69 \frac{1 \times 1.5 \times 2 \times 36}{4 \times 4.5 \times 6.25 \times 4.5} + .630 \frac{1 \times 1.5 \times 6.25 \times 36}{4 \times 4.5 \times 2 \times 4.5} + .2488 \frac{1 \times 4.5 \times 6.25 \times 36}{4 \times 1.5 \times 2 \times 4.5} + .00094 \frac{4 \times 4.5 \times 6.25 \times 36}{1 \times 1.5 \times 2 \times 4.5} \right)$$

$$= \frac{0.10}{.94} (1.000 + 1.312 + 4.67 + .382) = .793 \text{ lb-in}$$

Since $T_m = T_1 = 870$ lb-in (see Table 3), the torque increase required by accelerating the gear train is less than 0.1 percent and therefore may be neglected.

B. ELEVATING MECHANISM FOR FIRING CONDITION

45. Determine the gear train characteristics of the foregoing weapon during firing. The weapon is not elevating at this time so the motor need not accelerate against the reverse torque. Only the firing couple and the residual weight moment less the trunnion bearing frictional torque are applied. Friction helps to retard the reverse torque. From Paragraph 41

T_e = 26800 lb-in

T_b = 1500 lb-in

According to Equation 3, the firing couple is

$$T_f = aF_g - bF_a = 210{,}000 \text{ lb-in}$$

where a = 0.3 in

b = 0.2 in

F_g = 1,800,000 lb, max. propellant gas force

F_a = 1,650,000 lb, inertia force of recoiling parts

The reverse torque applied by the elevating arc is

$$T_E = T_f + T_e - T_b = 235{,}300 \text{ lb-in}$$

The reverse torque is less than the required accelerating torque therefore no change in gear sizes are necessary. Had the reverse torque been larger, it would have been necessary to make the teeth correspondingly larger to compensate for the increased load.

C. MANUAL ELEVATING MECHANISM

46. Determine the characteristics of a gear train equipped with worm and worm gear for handwheel operation if data is the same as the preceding examples. The schematic of the gear train is shown in Figure 17. Because accelerations are negligible during handwheel operation, only static resistances are to be overcome. From Paragraph 41

$$T_e = 26{,}800 \text{ lb-in}$$
$$T_b = 1{,}500 \text{ lb-in}$$
$$T_E = T_e + T_b = 28{,}300 \text{ lb-in}$$

The gear ratio, as suggested in Paragraph 29, is set at 640:1 The required handwheel torque based on 100% efficiency is

$$T_m = \frac{T_E}{r_g} = \frac{28300}{640} = 44.25 \text{ lb-in}$$

The losses in the meshes of the gears, worm wheel and worm will more than double this torque which alone is almost the ideal maximum of 50 lb-in. Although a 10-mil-elevation-per-turn of the handwheel provides a convenient count, half this count of 5-mil-per-turn will be only slightly less desirable. On this basis the gear train ratio will be 1280:1. The torque at the handwheel, excluding efficiency is

$$T_m = \frac{T_E}{r_g} = \frac{28300}{1280} = 22.1 \text{ lb-in}$$

The gear ratio of the worm and gear is 64:1, decreased from 1280:1 in two steps as shown schematically in Figure 17. The worm is irreversible, therefore it locks against any reverse torque. The reverse torque is assumed to be 235,300 lb-in, the same as that calculated in Paragraph 45.

$$T_E = 235{,}300 \text{ lb-in}$$

$$T_w = \frac{T_E}{r_{g1}} = 11{,}770 \text{ lb-in, torque on worm gear}$$

where $r_{g1} = 20$, the gear ratio between worm gear and elevating arc.

47. The design data of the worm gearing is obtained by following the procedures in Article 15-6 of Reference 6. The worm is malleable cast iron and the gear is phosphor bronze. As a trial measure, assume

$P_c = 0.625$ in, circular pitch of worm gear
$P_L = 0.625$ in, axial pitch of worm
$N_w = 1.0$, number of threads in worm
$N = 64$, number of teeth in worm wheel
$\beta = 14\text{-}1/2°$, pressure angle

Then

$$P_d = \frac{\pi}{P_c} = 5.025 \text{ teeth/in, diametral pitch}$$

$$D_p = \frac{N}{P_d} = 12.72 \text{ in, pitch diameter of gear}$$

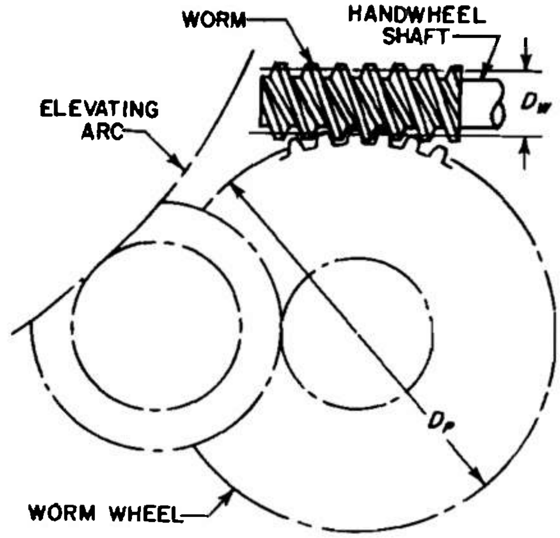


Figure 17. Gear Train, Manual Elevating

From page 386 of Reference 6

$D_w = 2.4\,P_L + 1.1 = 1.5 + 1.1 = 2.6$ in, pitch diameter of worm
$F_w = 2.38\,P_c + .25 = 1.49 + .25 = 1.74$ say 1.75 in, face width

The lead angle, λ, is 4° 22′ since

$$\tan \lambda = \frac{P_L\,N_w}{\pi\,D_w} = \frac{.625 \times 1.0}{2.6\pi} = .0765$$

According to Equations 15-17 and 15-18 of Reference 6, the limiting tooth load for wear is

$$F_{gw} = D_p F_w K_w = 12.72 \times 1.75 \times 150 = 3{,}340 \text{ lb}$$

and the limiting tooth load for beam strength is

$$F_{gs} = \sigma_e P_c F_w y = 24000 \times 0.625 \times 1.75 \times 0.1 = 2620 \text{ lb}$$

The values of K_w, σ_e, y are obtained from Table 15-2 of Reference 6.

$K_w = 150$ lb/in², wear factor
$\sigma_e = 24{,}000$ lb/in², endurance limit in bending
$y = 0.10$, Lewis factor

The gear load, excluding frictional effort is

$$F_g = \frac{T_w}{\frac{1}{2} D_p} = \frac{11770}{6.36} = 1{,}850 \text{ lb}$$

Subsequent calculations for a smaller and a larger circular pitch show the following results:

For $P_c = 0.5$, $F_{gw} = 2{,}300$ lb, $F_{gs} = 1{,}800$ lb, $F_g = 2{,}300$ lb
For $P_c = 0.75$, $F_{gw} = 4{,}500$ lb, $F_{gs} = 3{,}600$ lb, $F_g = 1{,}540$ lb

Neither of these is acceptable. The first has one of the allowable loads less than the actual tooth load. The second has too large a spread between the allowable loads and the actual loads. A gear with a higher strength-weight ratio may be designed but would require a special hob. If only a few gears are required, the additional cost to manufacture the special hob may not justify the small savings in weight and size.

48. The worm and gear are lubricated and protected from dirt to preserve a low coefficient of friction. However, low friction means a small lead angle is required if irreversibility is to be achieved. Reference 6 suggests assuming the coefficient of friction of 0.10. Since $\tan \lambda = 0.0765 < \mu = 0.10$, the worm cannot be driven by the gear and reverse torques will be locked out. With this required small lead angle, the efficiency of the worm gearing will be low.

From Equation 16

$$\eta_w = \frac{\cos \beta - \mu \tan \lambda}{\cos \beta + \mu \cot \lambda} = \frac{.968 - .10 \times .0765}{.968 + .10 \times 13.1} = .422$$

where $\beta = 14\text{-}1/2°$, pressure angle

If the friction in the thrust bearing of the worm is considered, then the efficiency according to Equation 17 is

$$\eta_w = \frac{\cos \beta - \mu \tan \lambda}{\cos \beta \left(1 + \mu_b \dfrac{D_b}{D_w} \cot \lambda\right) + \mu \cot \lambda \left(1 - \mu_b \dfrac{D_b}{D_w} \tan \lambda\right)}$$

$$= \frac{.968 - .10 \times .0765}{.968\left(1 + .01 \times \dfrac{1.5}{2.6} \times 13.1\right) + .10 \times 13.1 \left(1 - .01 \times \dfrac{1.5}{2.6} \times .0765\right)} = \frac{.960}{1.04 + 1.31} = .408$$

where D_b = 1.5 in, the effective diam. of the thrust bearing

μ_b = 0.01, assumed coefficient of friction of the thrust bearing

From Equation 18, the torque at the handwheel is

$$T_m = \frac{1}{\eta_w \eta_o^{n-1}} \frac{T_E}{r_o} = \frac{28300}{.408 \times .98 \times 1280} = 55.3 \text{ lb-in}$$

where η_o = .98 efficiency of each spur gear mesh

n = 2, number of spur gear meshes

This required handwheel torque is higher than the recommended value of 50 lb-in but is near enough to be acceptable (see Paragraph 31). The gear ratio of 1280:1 provides a convenient mil count at the handwheel, one revolution to 5 mils of elevation, and should be retained rather than strive for the optimum handwheel torque.

D. ELEVATING MECHANISM FOR MANEUVERING WEAPON

49. Determine the gear train characteristics for a typical weapon if, while traveling, it pitches at an angular acceleration of 55 rad/sec². The entire accelerating force to maintain the angular position between tipping parts and carriage is borne by the elevating mechanism. The motor is not required to develop this acceleration since a friction clutch near the motor checks the reverse torque. Firing does not take place, therefore only the unbalanced moment after equilibration and the frictional torque of the trunnion bearings combine with the torque induced by the pitching acceleration (see Paragraph 31). The known design parameters are

α = 0.5 rad/sec², normal elevating acceleration

α_p = 55 rad/sec², pitching acceleration

ω = 60°/sec, elevating speed

ω_m = 1800 rpm, motor speed

F_T = 10,000 lb, trunnion load

T_e = 5000 lb-in, unblanced equilibrator moment

T_f = 0, firing couple

r_b = 2.0 in, trunnion bearing radius

W_t = 1600 lb, weight of tipping parts

μ = 0.01, coefficient of friction

Φ = 1000 slug-ft², mass moment of inertia of tipping parts about the trunnion

From Equation 2

$$T_b = \mu F_T r_b = 0.01 \times 10{,}000 \times 2.0 = 200 \text{ lb-in}$$

The torque about the trunnions induced by the pitching acceleration has an additional component because the center of rotation is at the center of gravity of the vehicle rather than at that of the tipping parts. This component is influenced by the distance between trunnions and vehicle *CG* and, as in Figure 18, will have its maximum value when the elevation angle is such that the CG of the tipping parts, the trunnions, and the CG of the vehicle are colinear. At this time the accelerating force of the tipping parts is

$$F_a = \frac{W_t}{g} \alpha_p (R_v + r_t)$$

The induced torque is

$$T_a = \Phi_o \alpha_p + \frac{W_t}{g} \alpha_p (R_v + r_t) r_t$$

where Φ_o is the mass moment of inertia of the tipping parts about their own CG.
Since

$$\Phi = \Phi_o + \frac{W_t}{g}\frac{r^2}{t}$$

$$T_a = \left(\Phi + \frac{W_t}{g} R_v \frac{r}{t}\right) \alpha_p$$

$$T_a = \left(1000 \times 12 + \frac{1600}{386.4} \times 48 \times 24\right) 55$$
$$= 922{,}000 \text{ lb-in}$$

where R_v = 48 in, distance from trunnion to CG vehicle
r_t = 24 in, distance from trunnion to CG tipping parts
g = 386.4 in/sec^2, acceleration of gravity

From Equation 7

$$T_B = T_a - T_b + T_c = 926{,}800 \text{ lb-in}$$

Note that friction, although practically negligible, aids in resisting the reverse torque. Although the applied torque, in this instance, is treated as a static condition, the various gear ratios of the elevating mechanism are based on the specified elevating and motor speeds. The elevating angular velocity is equivalent to

$$\omega = 60°/\text{sec} = 10 \text{ rpm}$$

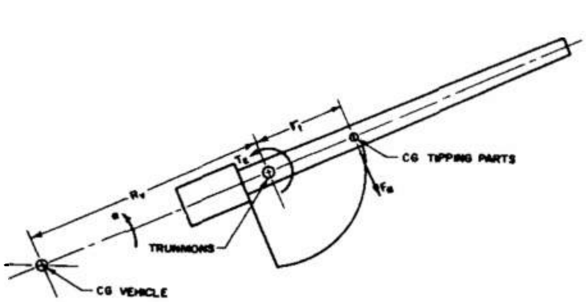


Figure 18. Geometry of Vehicle-Mounted Gun

The gear train ratio becomes

$$r_g = \frac{\omega_m}{\omega} = 180$$

One set of ratios which meet the gear train ratio is $r_g = 4 \times 3 \times 4 \times 3.75 = 180$, which can be met practicably by the individual gear sizes, so that

$$r_g = \frac{R_{p8}}{R_{p7}} \times \frac{R_{p6}}{R_{p5}} \times \frac{R_{p4}}{R_{p3}} \times \frac{R_{p2}}{R_{p1}} = \frac{24}{6} \times \frac{9}{3} \times \frac{6}{1.5} \times \frac{3.75}{1} = 180$$

The design data of the individual gears, based on the use of 20° ASA stub-tooth, are listed in Table 5.

TABLE 5. GEAR TRAIN DESIGN DATA, MANEUVERING MOUNT

Gear	R_p (in)	T (lb-in)	F_g (lb)	ω (rpm)	V_p (fpm)	c_v†
8	24	926,800	37,900	0	0	1.0
7	6	223,000	37,900	0	0	1.0
6	9	223,000	24,800	0	0	1.0
5	3	71,800	24,800	0	0	1.0
4	6	71,800	12,100	0	0	1.0
3	1.5	17,800	12,100	0	0	1.0
2	3.75	17,800	4,750	0	0	1.0
1	1	4,660	4,750	0	0	1.0

Gear	y'	N	P_c (in)	F_c (in)	C_i	F_w (in)
8	.00094	144	1.047	4.188	.568	7-3/8
7	.00094	36*	1.047	4.188	.568	7-3/8
6	.0025	63	.897	3.588	.810	4-7/16
5	.0025	21*	.897	3.588	.810	4-7/16
4	.0048	60	.628	2.512	.890	2-7/8
3	.0048	15	.628	2.512	.890	2-7/8
2	.0042	60	.393	1.572	.90	1-3/4
1	.0042	16	.393	1.572	.90	1-3/4

* The number of teeth indicated in Figure 16 and the corresponding diametral pitch do not conform to standard cutters, therefore, the next lower number of teeth which does conform to a standard pitch is selected.
† Velocity being considered to be zero, c_v is equal to unity.

50. Detailed calculations are shown for pinion No. 5. The applied torque now being a reverse torque, the friction helps to resist it, otherwise the calculations are similar as those in Paragraph 42. To meet the strength requirements and still maintain the values of the other parameters, C_F is set at 4.0, the upper limit for effective tooth loading.*

$$F_{56} = \eta_g \times \frac{T_B}{R_{p8}} \times \frac{R_{p7}}{R_{p6}} = .98 \times \frac{926{,}800}{24} \times \frac{6}{9}$$
$$= 24{,}800 \text{ lb, tooth load}$$

* Reference 6, page 334.

$$y_t' = \frac{F_g}{\sigma_s c_v C_F D_p^2} = \frac{24{,}800}{70{,}000 \times 1.0 \times 4.0 \times 36} = .0025$$

$N_5 = 21$, number of teeth (see note * under Table 5)

$P_d = N/D_p = 21/6 = 3.5$ teeth/in, diametral pitch

$N_6 = D_{p6} P_d = 18 \times 3.5 = 63$, no. of teeth on gear No. 6

$P_c = \pi/P_d = .897$ in, circular pitch

$F_e = C_F P_c = 4 \times .897 = 3.588$ in, effective face width

From Table 13-5 of Reference 6, for a face width between 2.0 and 18.0 in

$C_i = 0.885 - 0.0175\ F_w$, thus $C_i = .810$ (see Paragraph 43)

$$F_w = \frac{F_e}{C_i} = \frac{3.588}{.810} = 4.43 \text{ in, say 4-7/16 in face width}$$

51. The drive motor is not required to perform against the reverse torque but it must supply the additional power to operate the gear train. The additional effort in this case is not large but must be considered. The mass moments of inertia of the gears are obtained similarly to those of Table 4 (see Paragraph 44).

TABLE 6. MASS MOMENTS OF INERTIA OF GEARS, MANEUVERING MOUNT

Gear	R_p	R_p^4	F_w	Φ_z	Φ_{zz}
7	6	1296	7.375	11.07	44.32
6	9	6561	4.4375	33.25	
5	3	81	4.4375	.41	4.73
4	6	1296	2.875	4.32	
3	1.5	5.1	2.875	.017	.420
2	3.75	198	1.75	.403	
1	1	1	1.75	.00203	.00203

$$\Phi_z = \frac{1}{2} M R_p^2 = \frac{\delta\pi}{2g} R_p^4 F_w = .00116\ R_p^4 F_w, \text{ lb-in-sec}^2$$

where F_w = face width, in

R_p = pitch radius, in

$g = 386.4$ in/sec²

$\delta = .285$ lb/in³, density of steel

According to Paragraph 30, the drive motor must operate the gear train under the combined acceleration of pitching and normal elevation plus the effort normally necessary to accelerate the tipping parts.

$\alpha_e = \alpha + \alpha_p = 55 + 0.5 = 55.5$ rad/sec², equivalent acceleration of gear train

From Equation 21

$$T_\alpha = \frac{\alpha_e}{\eta_g^{\,n-1}}\left(\Phi_{67}\frac{R_{p1}\,R_{p3}\,R_{p5}\,R_{p8}}{R_{p2}\,R_{p4}\,R_{p6}\,R_{p7}} + \Phi_{45}\frac{R_{p1}\,R_{p3}\,R_{p6}\,R_{p8}}{R_{p2}\,R_{p4}\,R_{p5}\,R_{p7}} + \Phi_{23}\frac{R_{p1}\,R_{p4}\,R_{p6}\,R_{p8}}{R_{p2}\,R_{p3}\,R_{p5}\,R_{p7}} + \Phi_{1}\frac{R_{p2}\,R_{p4}\,R_{p6}\,R_{p8}}{R_{p1}\,R_{p3}\,R_{p5}\,R_{p7}}\right)$$

$$= \frac{55.5}{.98^3}\left(44.32\frac{1.0\times1.5\times3\times24}{3.75\times6\times9\times6} + 4.73\frac{1.0\times1.5\times9\times24}{3.75\times6\times3\times6} + .420\frac{1.0\times6\times9\times24}{3.75\times1.5\times3\times6} + .00203\frac{3.75\times6\times9\times24}{1.0\times1.5\times3\times6}\right)$$

$$= \frac{55.5}{.94}(3.94 + 3.78 + 5.38 + 0.37) = 79.5 \text{ lb-in}$$

During normal operation, the torque required to accelerate the tipping parts

$$T_a = \Phi\alpha = 1000 \times 12 \times 0.5 = 6000 \text{ lb-in}$$

where $\alpha = 0.5$ rad/sec², the normal elevating acceleration. The unbalanced equilibrator torque and the frictional torque of the trunnion bearings remain unchanged

$T_e = 5000$ lb-in

$T_b = 200$ lb-in

Thus $T_E = T_a + T_b + T_e = 11{,}200$ lb-in

The torque at the motor pinion needed to accelerate the tipping parts is

$$T_1 = \frac{T_E}{r_g \eta^n} = \frac{11{,}200}{180 \times .98^4} = 67.5 \text{ lb-in}$$

The total torque at the motor is

$$T_m = T_1 + T_\alpha = 147 \text{ lb-in}$$

The horsepower required is

$$HP = \frac{T_m \omega_m}{33000} = \frac{147}{12} \times \frac{2\pi\ 1800}{33000} = 4.2$$

The torque required of the motor to accelerate the gear train is too low to influence the gear design but this effort raises the requirements of the motor by 118 percent. A motor having a nominal rating of five horsepower will be satisfactory, but it also must be capable of delivering continuously a torque of 147 lb-in at the shaft at any speed, up to rated-in either direction.

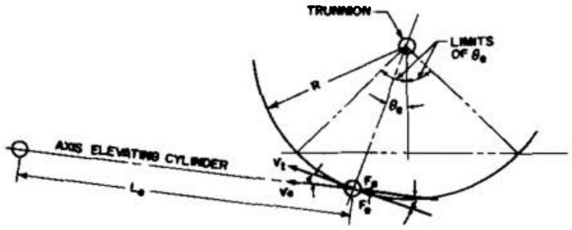


Figure 19. Geometry of Hydraulic Elevating Mechanism

E. HYDRAULIC ELEVATING MECHANISM

52. Figure 19 is the geometry of a hydraulic elevating mechanism to substitute for the unit in Problem D. All the design parameters are identical except that no gear train is involved. Elevation ranges from 65° to −5°. In this problem, maximum torque and maximum velocity may occur simultaneously but only during normal elevating operations. During high pitching accelerations, the unit merely supports the induced load. In Figure 19,

F_e = axial force of elevating cylinder
F'_e = effective force of elevating cylinder
L_e = length of elevating mechanism between pivots (variable)
R = radius from trunnion to pivot on cradle
v_e = velocity of elevating mechanism rod
v_t = tangential velocity at end of turning radius
θ_e = angular displacement of R from its mid position
ϵ = angle of F'_e and v_t with axis of elevating cylinder

Since the elevation range extends through 70°, the maximum value of θ_e is 35°. If $\epsilon = \theta_e$ at the extremes, the elevating mechanism is operating at its lowest required loads. Any deviation from this geometry will demand larger forces (F_e) through a considerable portion of the elevating range. For the pitching condition when T_E = 926,800 lb-in and if R = 15 in, F_e is maximum when $\epsilon = \theta_e$. When ϵ = 35°

$$F'_e = \frac{T_E}{R} = 61{,}800 \text{ lb}$$

$$F_e = \frac{F'_e}{\cos \epsilon} = \frac{61{,}800}{.819} = 75{,}400 \text{ lb}$$

If the ID of the cylinder and the diameter of the piston rod are 5.0 in and 2.0 in, respectively, the pressure area of the elevating piston is

$$A_e = \frac{\pi}{4}(5.0^2 - 2.0^2) = 16.5 \text{ in}^2$$

The maximum induced pressure in the cylinder is

$$p_e = \frac{F_e}{A_e} = 4560 \text{ lb/in}^2$$

During normal elevation,

$$\omega = 60°/\text{sec or } 1.048 \text{ rad/sec}$$

T_E = 11,200 lb-in. (computed in Paragraph 51)
Since one horsepower equals 6,600 in-lb per second, the required horsepower for elevation is

$$HP = \frac{T_E\omega}{6{,}600\,\eta} = \frac{11{,}200 \times 1.048}{6600 \times .90} = 1.98$$

where η = .90 (the assumed 90% efficiency of the hydraulic unit). This power requirement excludes the effort necessary to operate the moving components of the elevating mechanism which is highly likely to be less than that needed to activate a gear train of equivalent capacity. Since the applied torque is maintained constant, the horsepower required is the same regardless of the position of the elevating cylinder.

F. BRAKE

53. Determine the design data for an internal-shoe brake required to stop the elevating system of Chapter V, Part A after it reaches the maximum rated angular velocity. The brake is installed on the shaft connecting gear No. 2 and pinion No. 3 of Figure 15. Here the velocity is 187.5 rpm which is equivalent to 750 rpm at the motor shaft. The brake is designed for a stopping torque of 2-1/2 times the driving torque at gear unit No. 23. The procedures for computing the brake characteristics are those presented in Chapter 20 of Reference 6. Figure 20 is a sketch indicating the dimensions and loads of the brake.

The known data are

ω_{23} = 187.5 rpm, angular velocity of gear 2 & pinion 3
T_m = 870 lb-in, output torque of motor (Paragraph 44)

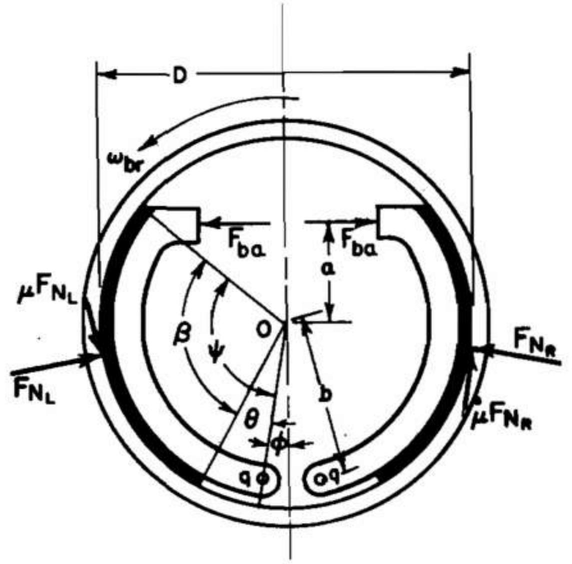


Figure 20. Brake With Applied Loads

$D = 2R = 16$ in, brake diameter at rubbing surfaces
$w = 1.5$ in, width of brake band
$\mu = 0.3$, coefficient of friction

Also known are these dimensions, (see Figure 20)
$a = 4.75$ in; $b = 7.0$ in; $\phi = 10°$; $\theta = 20°$; $\beta = 100°$; $\Psi = \beta + \theta = 120°$

Moments are taken about q, the pivot of the brake shoe, and consist of three components. These are computed according to Equations 20-11c, 20-12 and 20-13 of Reference 6 and apply to both left and right shoes. The sign of each component depends on the direction of rotation.

$$M_P = F_{ba}(a + b\cos\phi) = (4.75 + 7 \times .985)\,F_{ba} = 11.65\,F_{ba}$$

$$M_N = \frac{1}{4} p_m b\, w\, D\left(\Psi - \theta - \frac{1}{2}\sin 2\Psi + \frac{1}{2}\sin 2\theta\right)$$

$$= \frac{1}{4}\times 7\times 1.5\times 16\left(\frac{2\pi}{3} - \frac{\pi}{9} + \frac{1}{2}\times .866 + \frac{1}{2}\times .643\right) p_m = 105\,p_m$$

$$M_f = \frac{1}{4}\mu\, p_m w D\left[D(\cos\theta - \cos\Psi) - \frac{1}{2} b(\cos 2\theta - \cos 2\Psi)\right]$$

$$= \frac{1}{4}\times .3\times 1.5\times 16\left[16(.940 + .5) - \frac{1}{2}\times 7(.766 + .5)\right] p_m = 33.5\,p_m$$

where p_m = maximum pressure on the brake band. According to Equation 20-14 of Reference 6, the braking torque for the right shoe is

$$T_{bR} = \frac{1}{4} p_{mR}\,\mu D^2 w\,(\cos\theta - \cos\Psi)$$

$$= \frac{1}{4}\times .3\times 256\times 1.5\,(.940 + .50)\,p_{mR} = 41.4\,p_{mR}$$

Similarly for the left shoe

$$T_{bL} = 41.4\,p_{mL}$$

But, according to specifications, the total braking torque is required to be 2.5 T_{23}

$$T_b = 2.5 \times \frac{R_{p2}}{R_{p1}} T_m = 2.5 \times 4 \times 870 = 8700 \text{ lb-in}$$

The sum of the moments about the pivot of each shoe is equal to zero, therefore the total summation equals zero. If clockwise moments are positive, then

$$(M_N - M_f - M_P)_L + (-M_N - M_f + M_P)_R = 0$$

Since the brake shoes are symmetrical, $M_{PL} = M_{PR}$ and

$$M_{NL} - M_{fL} = M_{NR} + M_{fR}$$

Substituting the above values for M_N and M_f, we have

$$(105 - 33.5)\,p_{mL} = (105 + 33.5)\,p_{mR}$$

$$\frac{p_{mL}}{p_{mR}} = \frac{138.5}{71.5} = 1.94 \text{ or } p_{mL} = 1.94\,p_{mR}$$

The total brake torque is

$$T_b = T_{bL} + T_{bR} = 8{,}700 \text{ lb-in}$$

In terms of p_m

$$41.4\,(p_{mL} + p_{mR}) = 8{,}700 \text{ lb-in}$$

Substituting for p_{mL}

$$2.94\,p_{mR} = 210 \text{ lb/in}^2$$
$$p_{mR} = 71.5 \text{ lb/in}^2$$
$$p_{mL} = 138.5 \text{ lb/in}^2$$

Balancing the moments about the pivots at q,

$$M_{PL} = M_{NL} - M_{fL} = 71.5\,p_{mL} = 9{,}900 \text{ lb-in}$$
$$M_{PR} = M_{NR} + M_{fR} = 138.5\,p_{mR} = 9{,}900 \text{ lb-in}$$

$$F_{ba} = \frac{9{,}900}{11.65} = 849 \text{ lb, applied brake force}$$

54. The angular velocity of the brake is

$$\omega_{br} = \omega_{23} = 187.5\,\frac{2\pi}{60} = 19.62 \text{ rad/sec}$$

The surface area of each brake shoe is

$$A_b = wR\beta = 1.5 \times 8.0 \times \frac{100}{180}\pi = 20.95 \text{ in}^2$$

Maximum torque on a shoe is

$$T_{bL} = \frac{41.4}{12} p_{mL} = 478 \text{ lb-ft}$$

The energy rate is

$$E_r = T_{bL}\,\omega_{br} = 478 \times 19.6 = 9{,}370 \text{ ft-lb/sec}$$

The energy absorption rate is

$$E_a = \frac{E_r}{A_b} = \frac{9370}{20.95} = 447 \text{ ft-lb/sec-in}^2$$

According to Article 20-4 of Reference 6, this value is acceptable.

The energy absorbed by the brake is

$$E_b = \frac{1}{2}\Phi_e\,\omega_{br}^2 = 71,400 \text{ lb-in}$$

where Φ_e = 370.39 lb-in-sec², the effective mass moment of inertia at gear unit 23.

$$\Phi_e = \Phi\left(\frac{R_{p3}}{R_{p4}} \times \frac{R_{p5}}{R_{p6}} \times \frac{R_{p7}}{R_{p8}}\right)^2 + \Phi_{67}\left(\frac{R_{p3}}{R_{p4}} \times \frac{R_{p5}}{R_{p6}}\right)^2$$

$$+\Phi_{45}\left(\frac{R_{p3}}{R_{p4}}\right)^2 + \Phi_{23} + \Phi_1\left(\frac{R_{p2}}{R_{p1}}\right)^2$$

$$\Phi_e = 370 + 0.05 + 0.07 + 0.25 + 0.02$$
$$= 370.39 \text{ lb-in-sec}^2$$
$$\Phi = 174{,}000 \text{ lb-ft-sec}^2 \text{ (see Paragraph 41)}$$

For values of R_{px}, see Table 3.
For values of Φ_{xx}, see Table 4.

$$\theta_{br} = \frac{E_b}{T_b} = \frac{71400}{8700} = 8.21 \text{ rad, brake drum travel}$$

$$t_b = \frac{2\theta_{br}}{\omega_{br}} = \frac{16.42}{19.62} = .837 \text{ sec, braking time}$$

$$\alpha_{br} = \frac{\omega_{br}}{t_b} = \frac{19.62}{.837} = 23.4 \text{ rad/sec}^2\text{, deceleration at brake}$$

$$\alpha_{eb} = \alpha_b\frac{1}{r_{g23}} = \frac{23.4}{75} = .312 \text{ rad/sec}^2\text{, deceleration of tipping parts due to braking}$$

where r_{g23} = 75, the gear ratio of the train between tipping parts and brake

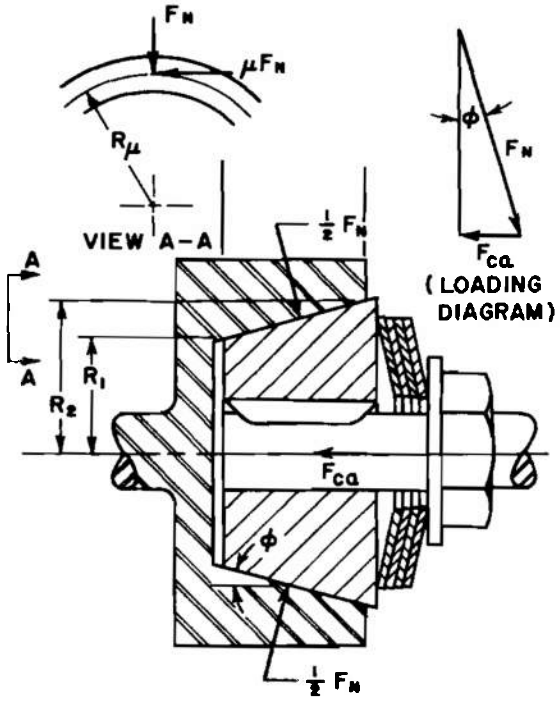


Figure 21. Clutch With Loading Diagram

G. SLIP CLUTCH

55. Compute the design data for a cone clutch which slips at 926,800 lb-in torque induced by 55 rad/sec² pitching acceleration of the carriage (see Paragraph 49). The clutch transmits the torque between pinion No. 1 and the motor shown in Figure 15. A sketch of the clutch, showing dimensions and loads, appears in Figure 21. The torque when slippage impends is

$$T_m = \frac{T_E}{r_g} = \frac{926{,}800}{180} = 5150 \text{ lb-in}$$

Other known data are

R_2 = 3 in, large radius of clutch
R_1 = 2.5 in, small radius of clutch
ϕ = 14° 55′, slope of conical surface
μ = .25, coefficient of friction

The radius of the friction circle is

$$R_\mu = \frac{2}{3}\left(\frac{R_2^3 - R_1^3}{R_2^2 - R_1^2}\right) = 2.76 \text{ in}$$

The torque transmitted by the clutch is

$$T_c = \mu F_N R_\mu$$

But

$$T_c = T_m = 5150 \text{ lb-in}$$

The total force normal to the surface is

$$F_N = \frac{T_c}{\mu R_\mu} = \frac{5150}{.25 \times 2.76} = 7470 \text{ lb}$$

From the loading diagram of Figure 21, the thrust component of the normal force is

$$F_{ca} = F_N \sin \phi = 7470 \times .2574 = 1920 \text{ lb}$$

The surface area of the clutch is

$$A_c = \frac{\pi}{\sin \phi} (R_2^2 - R_1^2) = 33.5 \text{ in}^2$$

The contact pressure of the clutch is

$$p_c = \frac{F_N}{A_c} = 223 \text{ lb/in}^2$$

56. The clutch should be completely enclosed for protection against dirt and weather so that variations in its frictional properties will be held to a minimum. It should be adjustable to the extent that variation in the coefficient of friction can be compensated for by variations in the axial thrust. The thrust is usually provided by a spring, either coil or Belleville, the latter being preferred because of its relatively small size and because large loads are available at small deflections. A typical installation is shown schematically in Figure 21. Calculations are shown for the design data of a Belleville spring which will provide the desired axial thrust. The equation and procedure are found in Reference 11.

d_i = 1.5 in, *ID* of spring
d_o = 3.5 in, *OD* of spring
t = 0.025 in, thickness of spring
h = 0.25 in, free height minus thickness
ν = 0.30, Poisson's ratio for steel

From curves on page 53 of Reference 11

$$C_1 = 1.28,\ C_2 = 1.48,\ M = 0.74$$

The stress at a deflection of Δ = .025 in is

$$\sigma = \frac{E\Delta}{(1 - \nu^2)\, M a^2}\left[C_1\left(h - \frac{\Delta}{2}\right) + C_2 t\right]$$

$$= \frac{29 \times 10^6 \times .025}{.91 \times .74 \times 3.06}\left[1.28(.25 - .0125) + 1.48 \times .025\right]$$

$$= 120{,}000 \text{ lb/in}^2$$

where

$$a = \frac{1}{2} d_o$$

The load which will produce this deflection is

$$F_s = \frac{E\Delta}{(1 - \nu^2)\, M a^2}\left[\left(h - \frac{\Delta}{2}\right)(h - \Delta)\, t + t^3\right],$$

per washer

$$= \frac{29 \times 10^6 \times .025}{.91 \times .74 \times 3.06}(.2375 \times .225 \times .025 + .00002)$$

$$= 478 \text{ lb/washer}$$

Four parallel washers will increase the spring load to F_s = 1912 lb, practically matching that of the design force of F_{ca} = 1900 lb. If necessary, spring loads can be changed by simply adjusting the deflection. Although greater deflections will increase the stress, the present stress of 120,000 lb/in² may reach 200,000 lb/in² before permanent set begins.

H. BUFFERS

57. Assume that the weapon in Sample Problem A is one limited in elevation and buffers are needed to stop the elevating system. The buffers are located 40 inches from the trunnion axis and have a stroke of 6 inches. They are designed to induce constant deceleration by absorbing the rotational energy at the maximum angular velocity of 15° per second while the torque of the drive motor is still applied. In Figure 14, R_b = 40 in, a = 10 in, and L_b = 4 in at the end of the buffer stroke.

Equation 32 may now be written

$$40 \sin \theta_{bm} + 4 \cos \theta_{bm} = 10$$

When put in terms of $\sin \theta_{bm}$ and reduced to its simplest terms

$$\sin^2 \theta_{bm} - .49506 \sin \theta_{bm} + .05198 = 0$$

Solving the quadratic equation and using the root that applies,

$$\sin \theta_{bm} = .15130$$

$$\theta_{bm} = 8° 42' = .152 \text{ rad}$$

From Equation 33

$$\alpha_b = \frac{\omega^2}{2\theta_{bm}} = \frac{.0684}{.304} = .225 \text{ rad/sec}^2$$

From Equation 37

$$v_b = 40\sqrt{.0684 - .45\,\theta_b} \text{ in/sec}$$

According to Equation 35

$$\Phi_s = 12 \times 174{,}000 + 4.69 \times 64 + .63 \times 625 + .2488 \times 5625 + .00094 \times 90{,}000 = 2{,}089{,}000 \text{ lb-in-sec}^2$$

The mass moments of inertia of the gears are in Table 4, of the tipping parts, in Paragraph 44.

From Equation 34

$$T_\alpha = 2{,}089{,}000 \times .225 = 470{,}000 \text{ lb-in}$$
$$T_a = 209{,}000 \text{ lb-in (see Paragraph 41)}$$

From Equation 36

$$T = T_\alpha + T_a = 679{,}000 \text{ lb-in}$$

According to Equation 31c

$$F_b = \frac{679{,}000}{40 - \dfrac{10\text{-}40 \sin \theta_b}{\cos \theta_b} \tan \theta_b}$$

For simplicity in the table of calculations, rewrite the equation twice to read

$$F_b = \frac{679{,}000}{40 - \dfrac{A}{\cos \theta_b} \tan \theta_b} = \frac{679{,}000}{40 - B}$$

Assume that twin buffers are used at each end to provide symmetrical loading. Each cylinder has a 2.5 in diameter and the rods 0.75 in diameter. The effective buffer piston area is

$$A_b = 2 \frac{\pi}{4} (2.5^2 - .75^2) = 8.93 \text{ in}^2$$

From Equation 26, the orifice area will be

$$a_o = \frac{v_b}{c_o} \sqrt{\frac{\rho A_b^3}{2F_b}} = \frac{.284 v_b}{\sqrt{F_b}}$$

where c_o = .60, orifice coefficient
$\rho = 8.5 \times 10^{-5}$ lb-sec^2/in^4, mass density of hydraulic fluid

TABLE 7. COMPUTED BUFFER DESIGN DATA

θ_b	0°	2.0°	4.0°	6.0°	7.5°	8.7°
θ_b (rad)	0	.0349	.0698	.1047	.1309	.152
$\sin \theta_b$	0	.0349	.0698	.1045	.1305	.1513
$\cos \theta_b$	1.0	.9994	.9986	.9945	.9914	.9885
$\tan \theta_b$	0	.0349	.0699	.1051	.1316	.1530
$40 \sin \theta_b$	0	1.396	2.790	4.181	5.220	6.042
A	10	8.604	7.210	5.819	4.780	3.958
B	0	.301	.505	.615	.636	.612
40-B	40	39.699	39.495	39.385	39.364	39.388
L_B (in)	10	8.609	7.228	5.851	4.822	4.0
x_b (in)	0	1.391	2.772	4.149	5.178	6.0
F_b (lb)	17,000	17,120	17,200	17,250	17,260	17,250
v_b (in/sec)	10.48	9.18	7.70	5.86	3.92	0.
a_o (in^2)	.0028	.0199	.0167	.0127	.0085	0

Each buffer has two grooves cut lengthwise into the inner cylinder wall, 180 degrees opposed. Each groove is 1/8-inch wide. The groove depth at each position is

$$d_g = \frac{1}{4} \frac{a_o}{1/8} = 2\, a_o$$

GLOSSARY

aim. Point or direct a weapon so that its missile is expected to strike the target.

antibacklash device. Mechanism which applies static torque in two directions in a gear train so as to provide positive tooth contact regardless of which direction the gears are to be turned.

buffer, elevating. Mechanism which absorbs the kinetic energy of the tipping parts of a weapon in elevation.

cannon. Component of a gun, howitzer, or mortar consisting of the complete assembly of tube, breech mechanism, firing mechanism or base cap.

carriage, gun. Mobile or fixed support for a cannon.

clutch, slip. Clutch designed to transmit a predetermined torque and to slip at a greater torque.

cradle. Nonrecoiling structure of a weapon which houses the recoiling parts and rotates about the trunnions to elevate the cannon or launcher.

depress. To decrease the angle of elevation.

director. Electromechanical equipment capable of computing pertinent firing data and then aiming the weapon.

double recoil gun. A weapon in which the gun recoils on the top carriage and the top carriage recoils on the bottom carriage.

elevating arc. Upright, geared arc attached to a weapon or carriage by which the weapon is elevated or depressed.

elevating cylinder. Cylinder which actuates a hydraulic elevating mechanism.

elevating mechanism. Mechanism on a gun carriage or launcher which elevates or depresses the weapon.

elevating mechanism, hydraulic. Elevating mechanism operated by hydraulic pressure which normally employs a cylinder, piston, and rod assembly which acts as a strut between carriage and cradle.

elevating mechanism, screw and nut type. Elevating mechanism which is activated by turning a screw or nut.

elevating system. All components involved in elevating or depressing a weapon.

elevation. Vertical angular position of a weapon with reference to the horizontal.

elevation, angle of. Vertical angle between the axis of the bore and the horizontal.

elevation, coarse. Placement of a cannon or launcher in the approximate elevation required by the problem.

elevation, fine. Precise positioning of a weapon in elevation.

equilibrator. Force-producing mechanism which provides a moment about the trunnions of a gun cradle or launcher which is equal and opposite to that caused by the unbalanced weight of the tipping parts.

equilibrator moment. Moment about the cradle trunnions which is produced by the equilibrator.

error signal. Signal in servomechanisms applied to the control circuit that indicates the misalignment between the controlling and controlled members.

fire control. Control over direction, volume, and time of fire of weapons.

firing couple. Couple about the trunnion axis generated by the resultant firing forces and the trunnion reaction.

firing cycle. Sequence of operation of a weapon from loading through firing.

friction circle. Circle on which frictional resistance to rotation may be considered to be concentrated.

gun. Piece of ordnance consisting essentially of a tube or barrel for launching projectiles by a force derived from an explosive.

hydraulic motor. Motor operated by hydraulic fluid under pressure.

hydraulic strut. Structural member consisting of cylinder, piston, and piston rod whose length can be varied by hydraulic pressure.

in-battery. A weapon is in battery when the gun tube has returned fully from recoil.

launcher. Device for supporting and holding in position for firing a rocket or missile.

laying. Act of directing or adjusting the aim of a weapon.

locking device. Fastening device which prevents inadvertent motion.

missile. Object that can be thrown, dropped, projected, or propelled for the purpose of striking a target.

mount, gun. Structure which supports a gun.

muzzle preponderance. Unbalance of the tipping parts of a weapon when the weight of the muzzle end exerts a greater moment about the trunnions than does the weight of the breech end.

No-Bak device. Device designed to deliver power at one end of a gear train turning in either direction but which will prevent a reverse torque at the other end from entering the original power source.

projectile. Missile fired from a gun.

propellant gas force. Force exerted on the base of a projectile by the propellant gases.

propellant gas period. Duration of the propellant gas activity.

pump, constant displacement. Hydraulic pump that has a constant stroke.

pump, variable displacement. Hydraulic pump whose flow is varied by varying the length of stroke.

range, elevating. Angular displacement to which a weapon is limited in elevation.

recoil. Movement of the gun tube and attached parts in direction opposite to the projectile travel.

recoil cycle. Complete sequence of recoil activity; in-battery, recoil, counterrecoil, buffing, in-battery.

recoil force. Total resistance to movement of the recoiling parts.

recoil mechanism. Unit which absorbs the energy of recoil.

recoil mechanism, concentric. Recoil mechanism which is concentric with the gun tube.

recoiling parts. Components of a weapon which move in recoil.

self-propelled weapon. Weapon incorporating its own prime mover.

shock absorber. Damper which moderates rapidly applied loads and prevents oscillation of the moving mass.

single recoil gun. Gun having only one complete recoiling unit.

stabilizer. Unit which tends to maintain the on-target position of the gun as a tank maneuvers.

tipping parts. Assembled structure of a weapon which rotates about the cradle trunnions.

top carriage. Upper structure of a gun carriage which supports the tipping parts and moves with the cradle in traverse.

torque, reverse. Torque induced by the maneuvering accelerations of a vehicle which tends to turn the gear train without the assistance of the power supply.

traverse. Horizontal angular displacement of a weapon in either direction.

traversing mechanism. Mechanism which turns the weapon in the horizontal plane.

trunnion. One of the two pivots supporting the cradle and recoiling parts on the carriage and forming the horizontal axis about which the piece elevates.

trunnion bearings. Bearings in which the trunnions rotate.

trunnion load. Load supported by the trunnions during any weapon activity.

valve, locking. Valve in a two line pressure system which permits flow in only one line at a time, meanwhile preventing flow toward the pressure source.

weapon. Instrument of combat.

weapon, mobile. Weapon which is readily transportable and therefore not restricted to one site for action.

weight moment. Moment about the trunnion axis caused by the weight of the tipping parts.

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